PITORIFICES AND SMALL PUMPS

IN COLD REGION

WATER DISTRIBUTION SYSTEMS

A

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By

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ABSTRACT

Most buried potable water distribution systems in colder regions of Alaska rely on pitorifices to provide circulation between the water main and service connections for freeze protection. Pitorifices are scoops which project into the main. When water is circulated in the main, they create a differential head which induces flow through dual service lines. Pitorifices have provided an inexpensive and simple alternative to installing a small pump at each service to provide circulation. However, very little information was available on the hydraulic performance of these devices.

The objectives of this study were to: (i) develop techniques to measure pitorifice performance in the field; (ii) characterize performance of commonly used pitorifice shapes with different insertion depths and relative sizes in full-scale testing; (iii) develop an improved shape; (iv) research the competing technology of small pumps; and (v) present the information in a way that is useful to engineers.

An inexpensive device for field checks of both differential head and flow rates at service lines was developed and the use of a low head loss meter was initiated. Methods and results of field studies in four different water systems are presented.

Five commonly used pitorifice shapes and four new shapes were evaluated. The best shape was found to be one of the existing shapes, which is also one of the easiest to produce but not the most popular. It was also determined using a larger service line size can be cost effective. Test results are graphed and a theoretical framework is provided for designers.

Smaller, energy efficient pumps may provide a cost effective alternative to pitorifices in some situations. Requirements for small pumps used for circulation in place of or to supplement pitorifices are given. Performance test results for different pumps are presented, most of which have not been used previously for service line circulation. Pumps with significantly lower operating costs than those in current use are identified. Several of these pumps were installed in services for long term testing.

Appendix G: Pitorifices and Small Pumps is a stand-alone summary of the most important parts of the thesis.

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NOMENCLATURE

A	Area, ampere, amperage
AC	Alternating current
BEP	Best efficiency point (of a pump)
BPM	Bypass manometer (See Subsection 5.2.5)
cs	Centistoke (measure of kinematic viscosity)
C	Correction factor, ampere · hour capacity of battery, Hazen-Williams pipe roughness factor
C_p	Specific heat, surface pressure coefficient
CUC	College Utilities Corp. (Fairbanks water utility)
D	Diameter
DI	Ductile iron (pipe)
E	Energy (heat, mechanical, etc.)
f	Friction factor (Darcy-Weisbach), frequency
F	Function
FC	Fluid column
fps	Feet per second
FPT	Female pipe thread
ftg	Fitting
g	Acceleration of gravity
gpm	Gallons per minute
h	Head, height, heat transfer coefficient
H	Head, height, depth of bury
HDPE	High density polyethylene
ID	Inside diameter
k	Thermal conductivity, constant
K	Correction factor, head loss or gain coefficient
kWh	Kilowatt hour
L	Length, latent heat
LF	Linear feet
LTW	Lighter than water (fluid, used in dual-fluid manometer)
$\dot{m}$	Mass flow rate
MPT	Male pipe thread

MUS	Municipal Utility Services (City of Fairbanks utility)
n	Rate of revolution
n_s	Specific speed of a pump (nondimensional number)
N_s	Specific speed of a pump (USCS units)
N_S	Number of services
Nu	Nusselt number (hD/k)
N_{Va}	Valensi number (Reynolds number of oscillatory flow)
OC	On-center (spacing)
OD	Outside diameter
P	Pressure
PHS	Public Health Service (U.S.)
PSC	Permanent split capacitor (type of AC motor)
PVC	Polyvinyl chloride
Q	Volumetric flow rate
r	Radial distance from center, radius measurement
R	Radius measurement, electrical resistance, thermal resistance
Re	Reynolds number ($Re = VD/\nu$)
rms	Root mean square
RPM	Revolutions per minute
SG	Specific gravity
SI	Système International (metric system of units)
SP	Shaded pole (type of AC motor)
St	Strouhal number (fD/V)
STP	Standard temperature and pressure (0°C, 760 mm Hg)
t	Time, thickness
T	Temperature
T_0	Period of oscillation
UAF	University of Alaska Fairbanks
USCS	United States Customary System (of units)
V	Velocity, voltage, volume
W	Weight, wattage
WC	Water column

Greek Symbols

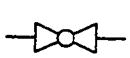
α	Kinetic energy correction factor for velocity profile in pipe
β	Bend angle, Bunsen coefficient
ϵ	Derived hydraulic (or equivalent) roughness of pipe
ϵ_0	Absolute geometric roughness of pipe
η	Efficiency
θ	Angle of flow measured perpendicular to a surface (degrees), fluid to solid contact angle
λ	Friction factor
μ	Absolute viscosity
ν	Kinematic viscosity
ϕ	Power factor
ρ	Density
Υ	Surface tension
ω	Natural frequency of a manometer fluid column

Subscripts

A	area, air
avg	average
BP	bypass
c	collar
C	conduit
d	demand
dn	downpit (downstream facing pitorifice)
D	drag
el	ell, elbow
eq	equivalent
f	friction, force, fitting
ftg	fitting
G	ground
GS	ground surface
i	impact (also pitot, stagnation or total) pressure. inside, insertion

I	insulation
LTWF	lighter than water fluid
m	mass, main, motor, mean
M	main
max	maximum
min	minimum
p	pressure, pitorifice, pitot tube, pump house, pump, pipe
o	outside. operating condition
O	oil
r	radius, return
s	service, sand, supply
S	series, soil, service
SC	service conduit
SO	shut-off
t	turbulent, tip, test condition, total
T	total, turbulent (zone of complete turbulence)
up	uppit (upstream facing pitorifice)
v	venturi
w	wall, water
w-w	wire-to-water (efficiency)
W	water

Symbols

	ball valve
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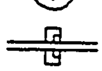
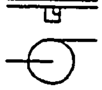
	globe valve
	differential pressure gauge or sensor
	temperature gauge or sensor
	union
	pump

UNITS

In a perfect world everyone would have been using a uniform and logical system of units from the start. The International System (SI, from the French "Le Système International d'Unité") of units is uniform and logical but it has not been in place long enough and the units are not uniformly applied. The standard SI unit of volumetric flow rate, derived from the basic units for length and time, is a cubic meter per second (m^3/s). Pump capacity, however, is usually given as cubic meters per hour to avoid the use of fractional numerical values for flow rate. The use of liter per second (L/s) is being promoted as a solution to this problem and it is used in this thesis. The standard SI unit of pressure is the Pascal (Pa), but it is so small that the bar (1 bar = 10^5 Pa) or the kiloPascal (kPa) are often used instead; the kiloPascal and meters of water column (m WC) are used in this thesis. In addition to these problems with SI units, there is still a large amount of literature which has used the older metric system with, for example, centimeters instead of millimeters or meters, centistokes instead of meters squared per second, and calories instead of Joules.

The United States Customary System (USCS) is also very much in use today and will have to be dealt with by engineers for a long time. Compromises with SI units include the use of conversion tables or presenting values in the text in both USCS and SI units with one of these in brackets, providing alternate scales on graphs, and providing two sets of tabled values.

The pipe and fittings used in this study were manufactured to dimensions given in USCS units and nominal sizes are customarily referred to using these units. Also, most previous related work has been done using USCS units, some of the equipment used in the study is calibrated in USCS units, and most of the engineers and water system operators who will use the results of this study are more familiar with USCS dimensions. For these reasons USCS units are often used in this thesis, particularly when referring to pipe, fittings, and previous work. Because SI units are much easier to work with, most data collection, data reduction, and calculations (except expample calculations) were done in SI units. Values are given in brackets in terms of alternate units in certain places where it was deemed helpful.

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CHAPTER 1: INTRODUCTION TO THE THESIS

A safe, reliable, convenient source of water is one of the most cost-effective public health measures that can be taken. In cold regions, potable water distribution is expensive to develop because mains and services are often looped, additional pumps and heating equipment are often used, and insulation or deeper burial is required. Water systems in cold regions are also more expensive to maintain because of higher pumping and heating costs. Fairbanks, Alaska, was one of the last large communities in the United States to have a piped public water system. Many small Alaskan communities are still without running water.

Circulating water distribution systems are the most common means of providing residential water service in Alaska and Canada where a conventional system would be subject to freezing because of permafrost or deep frost penetration. Water in the mains is kept from freezing by heat addition and pumped circulation. When sufficient heat is not added by pump and pipe friction losses and by heat conduction to the water at the service connections, additional heat may be added by mixing of warm make-up water or by passing some or all of the circulated water through a heating unit.

Water in the service lines is either bled to a drain, circulated back to the main, or heated electrically to keep it from freezing. Service line circulation in Alaska is usually by pitorifices and in Canada by small pumps. Pitorifices are scoops which project into the main and create a differential head to induce flow through dual service lines. Pitorifices were developed and used for the first time in 1953. Small circulation pumps were probably first used on service lines sometime in the late 1950s. Pitorifices require high circulation rates in the main which in turn cost Alaska communities several hundred thousand dollars a year in electricity. Small circulation pumps at the houses commonly use 80 watts year round; where electricity costs $0.45/kWh this represents a cost of over $300/year for the homeowner for a per capita cost several times higher than that for pumping costs in pitorifice systems.

1.1 HYPOTHESIS FOR THE STUDY

It was hypothesized that a more complete understanding of the functioning of pitorifices would lead to more economical designs and operating strategies for cold region water systems. An improved pitorifice design would provide either more head for circulation in the service lines or allow lower flows in the main. In the first case, higher mass flow rates would be possible and water could be maintained at a lower temperature with savings in heating costs possible. In the second case, considerable savings in pumping costs would be realized. A better understanding of pitorifice performance would also allow the engineer to better model a proposed system and to optimize the size of the lines, quantity of insulation, and the balance of heating and circulation costs.

It was also hypothesized that pumps commonly used in place of or in addition to pitorifices are oversized and inefficient. With the use of smaller, more efficient pumps, the cost advantage of pitorifices is diminished. At some point, the engineer should start using distributed pumping instead of pitorifices for lowest operating costs.

1.2 NEED FOR THE STUDY

Design of systems using pitorifices has historically been based on maintaining a two feet per second (fps) flow in the main and on using individual circulation pumps on services runs longer than about 60 feet. This flow velocity is based on the original 1953 Fairbanks design and, while it may be applicable for ¾-inch diameter service lines, it is unnecessarily high for 1-inch diameter service lines. This simplistic design approach ignores the fact that it is the heat loss rate and acceptable temperature drop in a service line which should dictate the flow in the service. If a line is well insulated, the flow in the service and hence the flow in the main may be reduced. This is particularly true if waste heat is available making it economical to use warmer water. About $120,000 is spent annually by the Fairbanks Municipal Utilities System on electricity for maintaining circulation in the mains. Pumping power requirements are proportional to the mean velocity in the main raised to the 2.85 power ($V_m^{2.85}$). Decreasing the velocity in the main by four percent results in an eleven percent reduction in energy requirements and about a seven percent decrease in service line flow rate. The cost of maintaining a given flow rate in a service can also be greatly reduced by going to a larger diameter service line. The additional cost of using 1-inch diameter service lines instead of ¾-inch diameter lines in new or replacement construction may be offset by the energy savings realized from being able to circulate water in the main at a lower rate. Finally, insulation, line size, and the relative consumption of electricity for pumping and fuel oil for heating can be optimized with a better understanding of their interdependence.

Prior to this study there was very little information available to the design engineer on the performance and economics of pitorifice systems. The only field test technique for pitorifice performance was the use of a manometer to measure the head differential at shut-off. Attempts had been made to measure flow rates using a rotameter or by observing injected dye, but these techniques have serious drawbacks. Heat loss monitoring had apparently never been done on installed service lines.

Figure 1.1a shows views of the original research model used by Page (1953): pitorifices with 0.75-inch outside diameters (ODs) were spaced 12 inches apart and oriented 90° relative to each other in a wood stave main with a 4-inch inside diameter (ID). Flows in the main were varied from 1.1 to 4.4 fps. Only 15 measurements were taken on this pitorifice design, twelve of which were with the tips of the pitorifices inserted to the center of the main, the other three with the tips withdrawn ½-inch from the center. Only a few data points had been obtained since then for other configurations and sizes, even though pitorifice ODs vary from 0.75-inch up to 2-inch with insertion depths of 2 to 3.5 inches in 3-inch to 10-inch and larger mains . Figures 1.1b and 1.1c show common extremes in pitorifice application and illustrate how much practice has diverted from the 1953 experimental model.

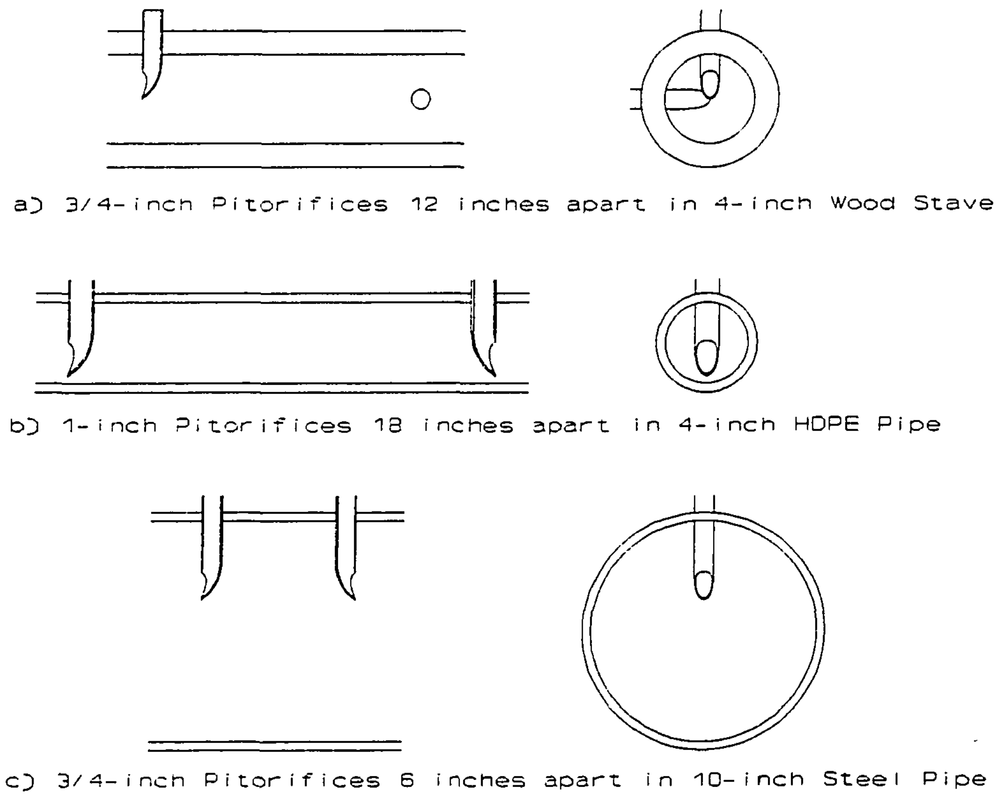


Figure 1.1: Pitorifices in Mains

If better design information and example calculations are available to engineers, more economical systems will be built. However, design information cannot be limited to just pitorifices. The major competing technology is the use of individual circulation pumps. Smaller, more efficient pumps have become available in recent years and they offer several advantages over pitorifices. Although not central to this study, the competition they offer as well as the possibilities of adopting them in a hybrid system need to be evaluated. However, no studies had been done on the contribution of motor heat to warming the water for the wet rotor types of pumps available. Also, most engineers are unaware of special plumbing requirements and the range of models available.

1.3 OBJECTIVES OF THE STUDY

The objectives of this study were to: (i) develop techniques to measure pitorifice performance in the field; (ii) characterize performance of commonly used pitorifice shapes with different insertion depths and relative sizes compared to the mains; (iii) develop an improved shape for pitorifices; (iv) research competing technologies and; (v) present the information in a way that is useful to engineers.

1.4 THESIS OUTLINE

Chapter 2 introduces the various types of cold region systems that have been developed and Chapter 3 reviews the development of pitorifices in Fairbanks. A theoretical basis for the study is presented in Chapter 4.

The requirements for test equipment, the selection and construction of equipment, and the general techniques used in the study are presented in Chapter 5. The characterization of pitorifice shapes, pitorifice construction, and the shapes used in the study are discussed in Chapter 6.

Test procedures, results, and conclusions for full and reduced scale testing of pitorifices are given in Chapter 7. Requirements for small circulation pumps, suitable offerings by various manufacturers, and pump test procedures, results, and conclusions are presented in Chapter 8. The procedures, results, and conclusions for field testing of pitorifices and small pumps are presented together in Chapter 9.

In Chapter 10, system analyses show how the data may be applied by an engineer. Finally, conclusions are summarized in Chapter 11.

Data from previous studies, most of it never published, are summarized in Appendix A. Additional data from this study are presented as graphs in Appendix B. Other topics of practical interest to engineers are covered in Appendices C through F. Appendix G is a stand-alone summary of those parts of the thesis of most interest to the practicing engineer.

In addition to the references given after the appendices, a separate bibliography is included which may be of interest to engineers working in areas related to the topics covered in this thesis.

CHAPTER 2: COLD REGION WATER DISTRIBUTION SYSTEMS

Over the past 90 years a number of systems and technologies have been suggested and developed. The engineer seeking the best solution for a particular community should be aware of all the various alternatives and their advantages and disadvantages. This chapter discusses different cold region water distribution systems in the order of their development, and then discusses elements common to all.

2.1 UTILIDORS, UTILIDUCTS, AND HEAT TRACING MAINS

Heated utilidors and direct heat tracing of buried pipe with steam or hot water are among the earliest techniques for freeze protection of water distribution systems. These systems were expensive to construct and usually limited in size. The downtown area of Fairbanks, Alaska was first serviced using these techniques in the early 1900s (Alter, 1977) and continues to be served this way today. Nenana, Alaska had a utilidor water system that served residents from 1914 to 1924 (Bainbridge, 1987). Winter watering points in small Alaskan communities are often supplied with above-ground or shallow-burial utiliducts. A utilidor system was constructed for a portion of Barrow, Alaska in the early 1980s (Leman, 1980; Martin and Sahlfeld, 1984; Zirjacks and Hwang, 1983). Service was extended with a buried pipe system because of the high cost of completing the utilidor system (Shillington and MacKinnon, 1987; Thomas, 1988). Gamble and Lukomskyj (1975) describes a number of different utilidor designs that have been used in Canada.

Direct electrical heat tracing of mains is occasionally done. In Greenland, some uncirculated distribution mains are kept just above freezing with electric heat trace. In Alaska, some circulating systems have heat trace as a back-up safety feature and for thawing.

2.2 BLEEDING WITH AND WITHOUT PUMP-BACK

The water system built in 1904 for Dawson City, Yukon Territory, is probably the earliest direct burial water distribution system in a region with permafrost. It consisted of shallow buried wood stave mains and relied on electrically heated makeup water supplied at 40°F, on bleeding service lines through ⅜-inch valves to drains, and on bleeding the looped main at the water plant (Anon., 1973; Dawson and Cronin, 1977; Engineer School, 1948; Grainge, 1958; Grainge, 1972; Reed, 1943; Yates and Stanley, 1963). Stanley (1965) reported that the system used surplus electrical power from a hydroelectric plant to heat the water. James (1980a) reported that the old wood stave was to be replaced with high density polyethylene, and Cameron (1992) reported it was replaced in 1980. The system is still operated as a bleed system with bleed rates of about 0.25 gpm. Figure 2.1 shows a schematic of a bleed system.

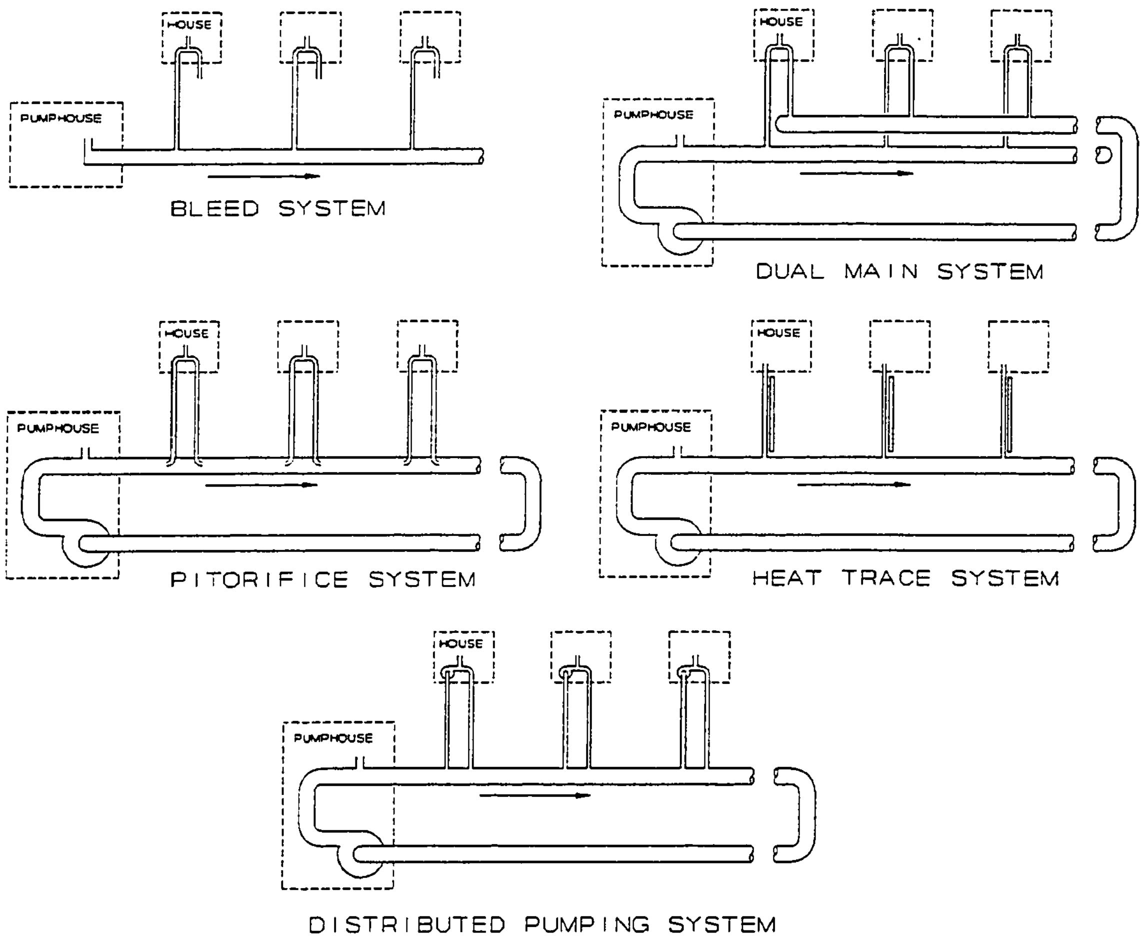


Figure 2.1: Schematics of Various Cold Region Water Distribution Systems

A major disadvantage of bleed systems is the high operating cost due to the wasting of heated, treated water. A solution to this is a bleed/pump-back device initially called "Resisto-Freeze" which was developed in 1982 in Faro, Yukon (Cotterill, 1983). The system is now called "Aqua-flo" and is manufactured by Powergain Manufacturing Ltd., Saskatoon, Saskatchewan. The system consists of a tank which is slowly allowed to fill and a float-operated pump which reinjects the water back into the service line. While the energy cost is low the capital cost is relatively high compared to pitorifice and distributed pumping systems.

A single line draw and return system using these bleed/pump-back devices was proposed as an option for a water distribution system in Mekoryuk, Alaska, but this was never built (Arctic Engineers Inc., 1983).

Bleeding has a secondary advantage of preventing sewers from freezing where water use is low (Stanley, 1965). Yee and Smith (1982) and Smith and Yee (1983) discuss bleeder control alternatives.

2.3 DUAL MAIN SYSTEMS

The first circulating system with looped service lines to individual homes was a dual main system built in the mining town of Flin Flon, Manitoba in 1931. Water in the mains was circulated at about 6 fps and heated to maintain a 38°F return water temperature. Mains were 4 to 16-inch diameter wood stave, 3 to 6-inch diameter cast iron, and 2-inch diameter wrought iron. Most of the lines were above ground and insulated with wood shavings and sawdust contained in wooden boxes; cinders were used for insulation when the lines were buried. Water flowed from a high pressure main through ½-inch diameter copper service lines back to a low pressure main. A thin copper disk with two or three ⅛ or $^3/_{32}$-inch diameter holes was inserted in a ½-inch flat-faced union downstream of the service line tee to limit the return flow to the low pressure main to around 1 gpm. When the freezing hazard was high, the flows in the service lines could be increased by increasing the pressure differential across the mains from 8 to 20 psi. Special fire hydrants were developed that would sit directly on the main. (Copp et al., 1956; Engineer School, 1948; McKay, 1982; Redman, 1950; Roche, 1948). This system has been extended and much of the above-ground pipe has been replaced with insulated buried pipe, but the system still is functioning as a dual main system.

A similar dual main system was constructed in Yellowknife, Northwest Territories in 1947 and 1948 (Copp et al., 1956; Grainge, 1959; Hall, 1951). In 1972 and 1973, after freezing problems were encountered in service lines due to a loss in pressure differential, the copper disks were removed and found to be enlarged. Stainless steel disks were used thereafter (Prentice and Srouji, 1980). Lock and Thierman (1974) reported that the Yellowknife system was particularly susceptible to imbalances caused by demand which made expansion as a dual main system difficult, so future expansion was as a single main circulating system. Dual main circulating systems are not being constructed anymore in Canada because of the past problems and the expense of installing two mains.

A dual main system was considered for Fairbanks in 1953 but rejected because of high capital cost. Collins and Jacobsen (1984) proposed a dual main design for McGrath, Alaska with both mains contained in the same insulated conduit to limit heat loss and with concurrent instead of countercurrent flows in the parallel mains to make flow balancing easier. A dual main system was built in Emmonak, Alaska in 1990 using flow balancing valves instead of orifices (Capito and Gajewski, 1991; Vause et al., 1987). A dual main system with counter flow is depicted in Figure 2.1.

2.4 SINGLE MAIN SYSTEMS WITH PITORIFICES

The forerunner of the pitorifice was the thermal tap proposed in 1948 by Amos Alter. The thermal tap consisted of upstream and downstream facing elbows inserted into a main with one located at the top and the other at the bottom of the main. The intent was to allow thermosyphoning due to temperature gradients in the main to induce flow in the service line during periods of minimum flow in the main, and to use the velocity head in the main to induce service line flow at other times (Alter, 1950a and 1991).

In 1952, Page, an engineer with the Arctic Health Research Center of the U.S. Public Health Service undertook exploratory research on a recirculating water system based on Alter's ideas (Murphy and Hartman, 1969; Page, 1954). This resulted in a design which required constant flow in the main and did not rely on a hypothetical thermosyphoning process. See Figure 2.1.

In 1953, R.W. Beck and Associates was asked to design a water distribution system for Fairbanks. A single main circulating system was chosen as the most cost-effective design and full-scale testing was done at Washington State University on various pitorifice designs. The term "pitorifice", a combination of the terms "pitot tube" and "orifice", was adopted by Westfall, an engineer with R.W. Beck, and Page (Westfall, 1990). It was decided that when the water in the mains was circulated at 2 to 3 fps, enough flow would be diverted through the ¾-inch diameter copper service lines to prevent freezing. Construction of the first phase was completed in late 1953 (Alter, 1969; Copp et al., 1956; Grainge, 1959; Page, 1953; Wallace and Westfall, 1954). Pitorifices used in the 1953 wood stave mains are shown in Figure 2.2. These were salvaged when the wood stave mains were replaced with ductile iron mains in 1991.

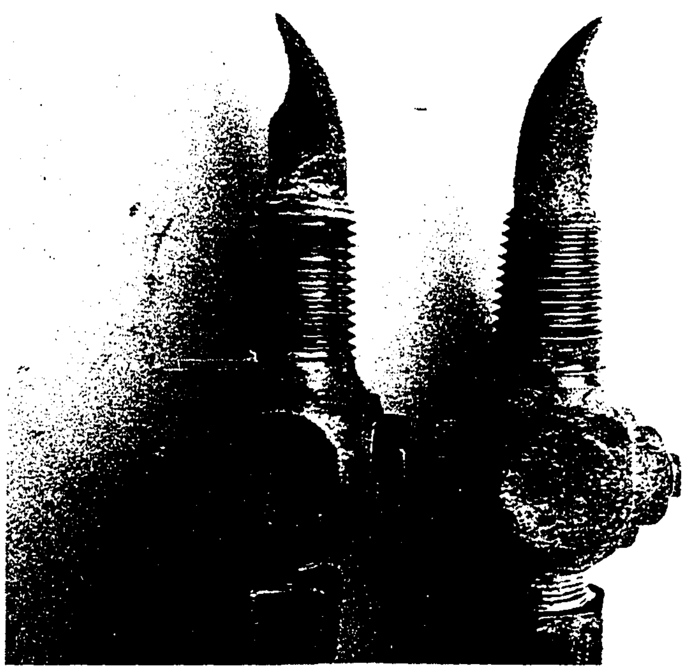


Figure 2.2: Original Pitorifices from Fairbanks' 1953 Wood Stave System

The Fairbanks design had the option of running pumps at a high speed to achieve 3 fps velocities. Because this option was seldom used, it was decided to design for 1.8 fps with a high speed option of 2.5 fps (Linck, 1961). The lower rate of circulation was not noted in many subsequent reports and the original high rate of 3 fps has been frequently reported in the literature. In fact, Grainge (1958) reported 3.5 fps was required and Lawrence (1969) reported 4.0 fps was required. An investigation by Johnson (1978) suggested that flow rates down to 1.0 fps should be feasible with 1-inch diameter service lines.

In 1965, single main circulating systems using pitorifices were installed in Nome and Unalakleet, Alaska, and shortly afterward in Grayling, Alaska (Alter, 1969; Leman et at., 1978; Murphy and Hartman, 1969). Water in Unalakleet was circulated at 1.7 fps with ½-inch diameter service lines (Ryan and Lauster, 1966). All these systems used pitorifices for residential services.

After the success of the single main circulating system in Fairbanks, the use of such systems was promoted (Clark and Alter, 1956; Copp et al., 1956; Dickens, 1959). While Canadians started using single main circulating systems, they have not used pitorifices very often. Pitorifices were considered for use in the single main circulating system built in Uranium City, Saskatchewan in 1957 and 1958, but an electric heating system was chosen instead (Klassen, 1960, 1965a, and 1965b). Pitorifices and individual circulation pumps were both considered for Resolute Bay, Northwest Territories (Dawson and Cronin, 1977). A pitorifice system was installed in 1978, but individual circulation pumps were added about eight years later due to freeze problems from poor construction and low circulation (Tam, 1992). Porter Creek, Yukon Territories (identified as a satellite of Whitehorse) was also reported to have used pitorifices (Lawrence, 1969).

The communities of Leadville and Grandby, Colorado used pitorifices but relied on bleeding or consumer use to provide flow in the mains rather than pumping (Wright and Fricke, 1963). Santori (1976) mentioned that Colorado utilities sometimes run double service lines with a pitot tube on one. These communities now rely on deep burial of mains and services for freeze protection.

2.5 SINGLE MAIN SYSTEMS WITH HEAT TRACING

Alter (1950b) described the use of nickel-chromium wire with an air thermostat to protect service lines, but he did not indicate where, or if, such an installation existed. The first use of heat tracing for a community system may be the cast-iron single main circulating system built in Uranium City, Saskatchewan in 1957 and 1958. The ¾-inch diameter copper water service line was laid on top of the sewer lines and both were insulated with a vermiculite asphalt mixture. Heating was achieved through the use of insulated wire which ran to each end of the copper service line and connected to a small transformer (Ellis, 1962; Klassen, 1960, 1965a, and 1965b). This type of heat tracing is described in detail by Eaton (1964). A system almost identical to Uranium City's was built a few years later in Thompson, Manitoba except that preformed polystyrene insulation was used to insulate service lines (Klassen, 1965a; Klassen, 1965b). Figure 2.1 shows a schematic of a heat trace system.

Electric heat trace is usually used with extended main systems. In extended main systems the main is brought very near the house and the service line length is very short. This type of system can have property access problems and is usually very costly because the mains are longer.

Self-limiting heat trace is also frequently used for backup freeze protection and for thawing high density polyethylene (HDPE) plastic service lines contained in insulated utiliducts. In general, however, heat tracing usually costs too much to install and to operate as a primary means of freeze protection.

Rosendahl (1980) reported heat trace was used inside high density polyethylene in Greenland. Internal heat tracing of high density polyethylene pipe was investigated by Mace (1984, 1985, and 1987). The internal heat tracing of a transmission line is also described by Cheriton (1966). Eaton (1964) showed internal heat tracing of the discharge pipe from a submersible well pump (well drop pipe).

Running sewer and water lines together to conserve heat can be considered a type of heat tracing. This can be safely done with modern materials and appropriate care, but has been a source of problems in the past. Alter (1977) describes the disastrous leaks caused in water mains in Palmer where they followed the sewer mains through the manholes and were exposed to corrosive conditions. James (1976) describes insulated service line bundles used in Canada. This type of construction can also result in lower construction costs.

2.6 SINGLE MAIN SYSTEMS WITH DISTRIBUTED PUMPING

Alter (1950a) wrote one of the first thorough reviews of cold region water distribution systems. He mentioned heat tracing and dual main systems, and proposed the single main circulating system. He did not, however, mention the use of individual circulation pumps. Billings (1953) suggested the use of individual circulation pumps but did not mention any location where they were being used. Perhaps the first use of such pumps was in Fairbanks where Linck (1961) reported they had been added on a few lengthy service lines to remedy freezing problems. Figure 2.1 shows a schematic of a distributed pumping system using small circulating pumps in each house.

Individual circulation pumps are occasionally used on service lines in Alaska but are not in general use because of their mechanical problems and high operating and replacement costs. Murphy and Hartman (1969) noted that requiring each customer to install and maintain a pump is not the most economical manner for the distribution of water, particularly in a large city. Individual circulation pumps were recommended in Nome after modifications to the main resulted in low flows (Corwin and Kniefel, 1983). They are also used in Barrow where the utility provides them and pays for their operation and maintenance (Shillington and MacKinnon, 1987; Thomas, 1988; Zirjacks and Hwang, 1983). In Huslia, Alaska individual circulation pumps are used as the primary means of circulating water in service loops. A high-speed pump at the pump house is operated in the event of a community power failure, and water is thereby circulated fast enough to allow the pitorifices to work effectively. Most other communities with single main circulating systems in Alaska use pitorifices, not circulation pumps, as the primary means of protecting service lines.

In 1981, the Alaska Area Native Health Service did tests to determine whether individual circulation pumps should be used in place of pitorifices (Carrubba, 1981). Test results recommended the continued use of pitorifices as the main means of circulation and an increase in service line size from ¾-inch to 1-inch diameter (Gerlek, 1982a; Gerlek, 1982b).

Individual circulation pumps are widely used in Canada. The reason for the difference between Canadian and Alaskan practice may be due both to precedents and construction funding practices. Where electric power is relatively cheap and reliable, the expense of maintenance and operation of a small pump is not as much as it would be otherwise, and where the individual consumer must pay for the service connection, the use of individual circulation pumps may be preferred by the agency funding the water system. If smaller and more energy-efficient pumps are used, it becomes more practical to use them in place of pitorifices.

2.7 SINGLE MAIN SYSTEMS WITH HEAD LOSS OR VENTURI DEVICES

Billings (1953) and Lawrence (1974a) suggested the use of orifices in the main to create a head loss which would induce flow through the service lines. Lawrence (1974b) also suggested the use of a venturi section in the main. Orifices in the main were considered for use in Saint Marys, Alaska (Mather, 1972a and 1972b) but pitorifices were used instead. The use of an orifice or venturi could allow less costly operation but the installation would be more expensive, particularly in the case of venturi devices. Fire flows would also be impaired. Cameron et al. (1977) and Rosendahl (1980) suggest alternatives to using the water system for fire fighting. Alternatives such as airport-type foam trucks could eliminate the need for fire flows.

2.8 SINGLE MAIN SYSTEM WITH INTERMITTENT USE

Murphy and Hartman (1969) suggested using intermittent pumping to fill tanks in houses with all lines drained between pumpings. A similar system, termed a pulsed water supply system, was proposed for Canadian communities (James, 1976; James and Robinson, 1979). Such a system has never been built in a cold region but should have low capital and operating costs. Water is supplied to holding tanks in homes through heat traced, small diameter mains. Air is used to clear the lines between deliveries and the heat trace is turned on shortly before the lines are to be used.

2.9 ELECTRICAL RESISTANCE, HOT WATER, AND STEAM THAWING

Shepperd (1934), Amsbary (1936), Bohlander (1963) and Nelson (1976) discuss electrical resistance methods of thawing mains and service lines, and Santori (1976) describes conductivity testing. Electrical conductivity in the main is ensured by welding pairs of straps across the joints and testing to ensure good connections. Electrical conductivity in copper service lines is ensured by using flare fittings and not pack joint fittings. Soldered joints are not permitted because they may fail during electrical thawing. Electrical thawing must be done with low voltage for safety reasons. Short circuits are a problem and piping must be electrically isolated before thawing is attempted.

Hot water has also been used to thaw service lines (Currey, 1980; Nelson, 1976). Western Water (Calgary, Alberta) markets the PAT 75 Power Thawing System which is used to circulate hot water inside service lines. Steam thawing is also used for thawing, but can cause damage to plastic pipe.

2.10 PIPE MATERIALS

Wood stave pipe was one of the first materials used because it has some inherent insulating properties. Polyvinyl chloride (PVC) pipe was used for many years in remote communities in Alaska because it was light and relatively cheap. However, it is no longer used because it is susceptible to freeze damage. High density polyethylene (HDPE) pipe may have first been used for a cold region water system in 1960 in Sweden (Janson, 1963). It is now standard in rural communities in Canada and Alaska for both mains and service lines because it withstands freeze damage so well. Ductile iron mains and soft annealed Type K copper water tube service lines are currently used in Fairbanks. Type K copper water tube has the thickest wall of the three common types. See Appendix F for more information on service lines.

2.11 PIPE INSULATION

Wood shavings, sawdust, cinders, and moss were the first insulating materials used for water lines. Vermiculite asphalt was used in 1953 in Fairbanks, and later in 1957, in Uranium City. Preformed polystyrene was used in Thompson a few years later. Field-applied polyurethane was considered for Seldovia, Alaska, but preinsulated pipe was used instead (Nyman, 1965). The first field foaming may have been for an insulated shallow-bury water line pipe fabricated on site at Elsa, Yukon Territories in 1965 (Anon., 1966). It was reported in 1968 that water mains in Fairbanks were being field foamed with blown foam insulation which was presumably urethane (Cornell, 1968).

A $^{9}/_{16}$-inch thickness (Smith (1976) reported this as a 1-inch thickness) of tar impregnated asbestos felt was used to protect and insulate steel water mains installed in 1955, in Fairbanks (Linck, 1961; Westfall, 1958). These lines may have been the first prefabricated insulated pipes used for water distribution. Steel pipe insulated with polyurethane inside cardboard or metal culverts, depending on burial depth, was installed in 1964, in Seldovia (Nyman, 1965). Preformed foamed glass (Foamglass®) was used in 1965 in Unalakleet, Alaska, on both the ductile iron mains and the copper service lines. Powdered insulation (presumably powdered and treated calcium carbonate) was used where the service lines tapped the main (Murphy and Hartman, 1969).

Lawrence (1969) mentioned that the "liberal use of non-absorbent insulations such as urethane foam and Styrofoam® is one of the most significant trends in buried utility installations in the North." Field-foamed polyurethane is currently specified for insulating services in Fairbanks. Foam specimens from installations up to 14 years old have shown negligible deterioration (Haigh, 1986). Polyurethane is also the preeminent insulation material for pre-insulated piping. Polyurethane foam can absorb water and if it does its insulating value will be lowered (Kaplar, 1978; McFadden, 1988).

CHAPTER 3: DEVELOPMENT OF THE PITORIFICE SYSTEM

Prior to 1953, water service in Fairbanks was limited to the downtown business area which was also served by a district heating system. Water lines were either in heated utilidors or utiliducts, or buried in close proximity to steam lines. In 1944, Black and Veatch of Kansas City designed a bleed type water system for Fairbanks. Eighty percent of the funding was to come from the Federal Works Agency. The project was terminated shortly after it was initiated when the government withdrew funding because the project was no longer needed to support the war effort (Anon., 1944; Nordale, 1945).

The concept of using upstream and downstream facing tubes in a water main to induce flow in service lines was first described by Alter. Alter had used Cole pitometers in 1939 to measure stream flow. In 1946, when he was stationed in Fairbanks as a commissioned officer in the U.S. Public Health Service, he started working on ways to deliver water in cold regions. He suggested putting upstream and downstream facing elbows at the top and bottom of a water main. The intent was to allow thermosyphoning to induce flow in the service line during periods of minimum flow in the main, and to use the velocity head in the main to induce service line flow at other times (Alter, 1950a and 1991).

In 1952, Page started research on a circulating water system based on Alter's ideas, using 2-inch diameter Lucite pipe and 0.5-inch diameter glass tubing to simulate the main and service lines (Murphy and Hartman, 1969; Page, 1952 and 1954).

3.1 EARLY FULL SCALE TESTING

In 1953, R. W. Beck and Associates of Seattle, Washington was asked to design a water distribution system for Fairbanks. Four systems were initially considered: single main with pitorifices, dual main, steam traced, and utilidor. The single main system was chosen because it had the lowest estimated capital cost (Wallace and Westfall, 1954). In support of the design, full-scale testing was done at Washington State College (now Washington State University) on various pitorifice designs.

Pitorifices were reported to have outside diameters (ODs) of 0.75 inch so they may have been fashioned from nominal ⅝-inch diameter copper water tube which has exactly that OD. Twelve pitorifice models were used in 20 different configurations for a total of 65 test runs. (Figure A.2 in Appendix A illustrates these shapes.)

A 60-foot length of 4-inch diameter wood stave pipe was used for the test main. Pitorifices were inserted one foot apart and at right angles to one another. It was not explained why pitorifices were not installed in the same plane. It may have been due to concerns about weakening the staves although they were installed in the same plane for the final installation in Fairbanks. Pitorifices were inserted in the wood stave pipe using brass tapping nipples with packing nuts. Rubber hose was used to connect the pitorifices to 20-foot lengths of copper water tube as shown in Figure 3.1. A dual fluid kerosine/water manometer was made from $^5/_{16}$-inch diameter plastic tubing but it was not noted whether this was ID or OD or what type of plastic was used. The copper water tube service line was reported to have an ID of 0.79 inch so it is presumed to have been nominal ¾-inch diameter Type L copper water tube which has an ID of 0.785 inch.

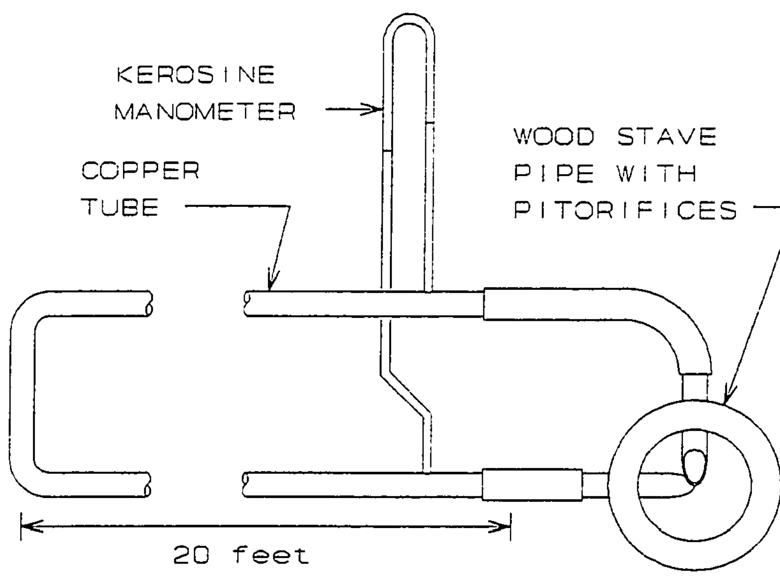


Figure 3.1: Schematic for 1953 Pitorifice Tests

Flow rates in the main were determined with a weighing tank. Flow rates in the service line were estimated using the water/kerosene manometer and a pressure drop versus flow relationship for the service line. Water temperature was reported to be 58°F although the viscosity used in calculations was for water at 68°F.

The induced flow velocity in the service was plotted as a function of the velocity in the main for a number of different pitorifice shapes and combinations. The tips of the pitorifices were inserted to the center line of the main in most tests. Using different shapes for upstream and downstream pitorifices was recognized as being more efficient but believed to be too prone to misapplication in the field. Although model B-2 was not the most efficient, it performed well and was easy to make and install, so it was chosen for the Fairbanks water system. Model B-2 is referred to in this thesis as a bend, cut and turn (BC&T) shape and is described further in Chapter 6. Data from the 1953 test are plotted in Figure 3.2, and data from the 1953 test and other tests are presented in Appendix A.

The induced flows for 100 and 200-foot total lengths of service line were then estimated by

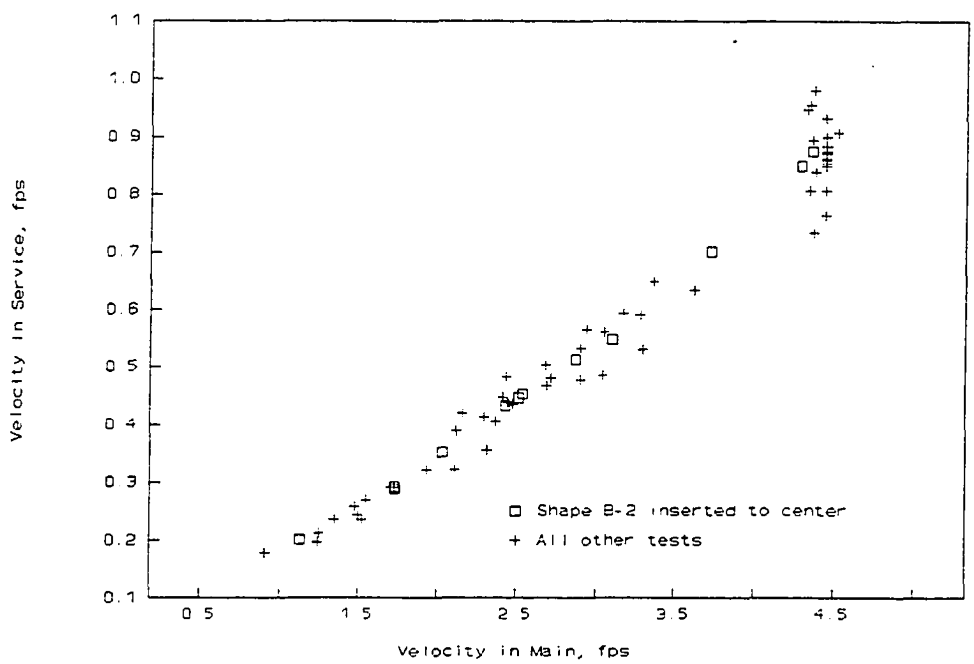


Figure 3.2: Pitorifice Performance Data from 1953 Study

extrapolating the data from the 40-foot test lengths. This was done by first determining the Darcy friction factor of the test length of service line:

$$f=\frac{2gD_sH}{LV_s^2} \tag{3.1}$$

where f is the Darcy friction factor, g is the acceleration of gravity, D_s is the service line ID, H is the head difference measured by the manometer, L is the length of service line, and V_s is the velocity in the service line. V_s was found from a previously determined calibration curve relating it to H. The ratio H/f was then calculated for every data point taken and used in turn to calculate V_s for L=100 feet and L=200 feet. Essentially, this amounted to assuming $L_1V_1^2=L_2V_2^2$ and contained the implicit assumption that either the friction factor of the service line and the head available for a given velocity in the main remained constant or that the ratio of the two remained constant over the range of extrapolation. However, none of the induced flows were in the fully turbulent region where the friction factor is independent of flow rate, and lengthening the service would result in even lower flow rates. In addition, many of the service line flows had Reynolds numbers in the 2,000 to 4,000 range which is the critical zone where flows transition between laminar and incompletely turbulent. Lengthening the service further would result in flows becoming laminar where the friction factor is inversely proportional to the velocity.

Assuming the friction factor remains constant also ignores the effect of entrance head loss. The pressure taps were located near the main and part of the entrance head loss may have been included. Figure 3.3 shows a plot of the calibration data used by Page and the theoretical head loss. Entrance effects may explain why the service line calibration curves do not follow theory, but the small diameter of the manometer tubing and changes in the contact angle between the manometer fluids and the inside surface of the manometer tubing or in the surface tension of the fluids over time may also account for some of the discrepancy (see subsection 5.1.1). It may also be that the calibration curve was obtained by disconnecting the rubber hose from the downstream pitorifice and making measurements while running water through the main. This would have resulted in the entrance losses being included, but there was no mention of this in the report. Also, no mention was made of how the specific gravity of the kerosine was determined. Apparently it was assumed to be 0.8 because the differential head was divided by five to convert to water column.

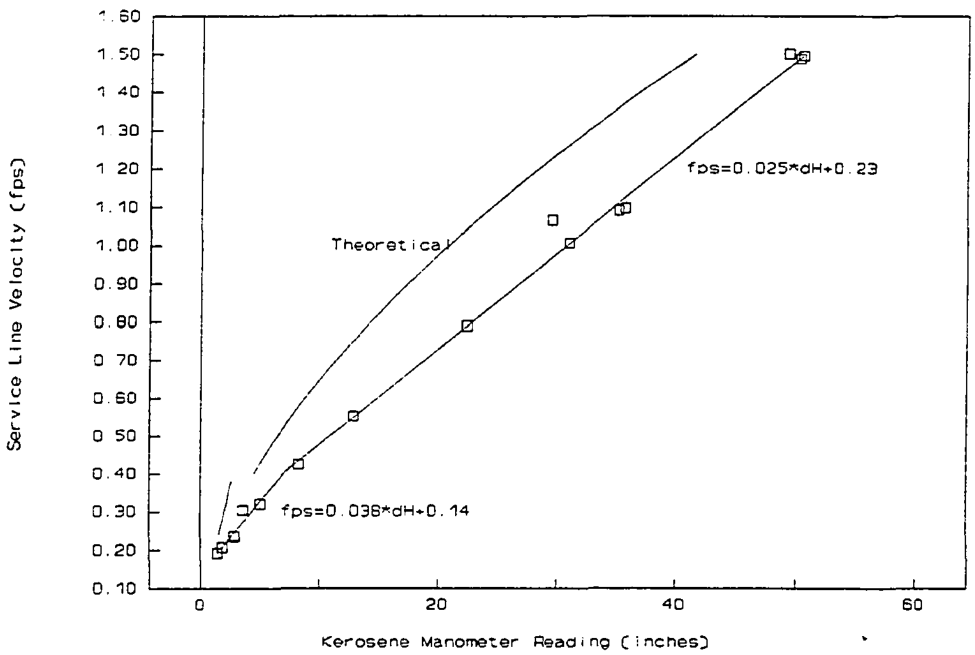


Figure 3.3: 1953 Manometer Calibration Data Compared to Theory

The implicit assumption that the head available across the pitorifices remains constant regardless of the flow in the service is also not strictly valid. The head at the opening of the upstream pitorifice will decrease as the difference in the squares of the main and service line velocities, following Bernoulli's equation, but the effect is not very significant because service line velocities will typically be much smaller than main velocities. The head developed at the downstream pitorifice by the turbulent wake may, however, be altered significantly as the service line flow rate changes.

Another implicit assumption was that either water would be circulated in the mains at 58°F or that the effects of temperature on the performance would be negligible. This last assumption is not valid because the Reynolds number, and hence the friction factor, is dependent on the kinematic viscosity of water which increases with decreasing temperature. In the case of laminar flow there is a direct inverse relationship between viscosity and velocity. Cooling the water from 58 to 40°F will result in increasing the viscosity by 33 percent and reducing flow by 25 percent when flow is laminar with the head constant for both cases.

Full-scale cold room tests were also conducted to determine the minimum service line flow needed to prevent freezing in uninsulated copper or plastic service lines. (Page did not identify the plastic but did say it could withstand freeze cycling.) Westfall and Wallace (1953) reported that:

> Two types of pipe were considered for use in the service connections ... copper and plastic. In making our analysis of the heat transfer of the service connections, we were unable to find any satisfactory basic data covering pipes of these materials buried in soil. Because of this lack of basic data and because of the special conditions involved in these connections, we determined to set up full-scale models of the connections and determine experimentally the data which could be used to properly determine the heat loss from the service connections.

For these tests, four ten-foot lengths of hard drawn copper water tube connected by sections of rubber hose were bedded in saturated sand in a cold room. Two of the sections were laid next to one another, and two were laid separate as shown in Figure 3.4.

Water was circulated through the pipe at various rates and temperature readings were made using thermocouples. The thermal conductivity was then estimated by noting that for small changes in water temperature relative to the temperature differential $T_w - T_s$:

$$\frac{\dot{m} C_p \Delta T_w}{L} = \frac{2 \pi k (T_w - T_s)}{\ln(r_s / r_w)} \tag{3.2}$$

where $\dot{m}$ is the mass flow rate, C_p is the heat capacity, ΔT_w is the temperature change of the water and T_w is the average temperature of the water, L is the total length of pipe, k is the thermal conductivity of the sand/water/ice system, r_s is the radial distance to the thermocouples in the soil from the center of the pipe, r_p is the pipe radius, and T_s is the soil temperature at r_s. T_s was assumed to be the average of four readings and pipe wall and water film thermal resistances where implicitly considered negligible. It was assumed

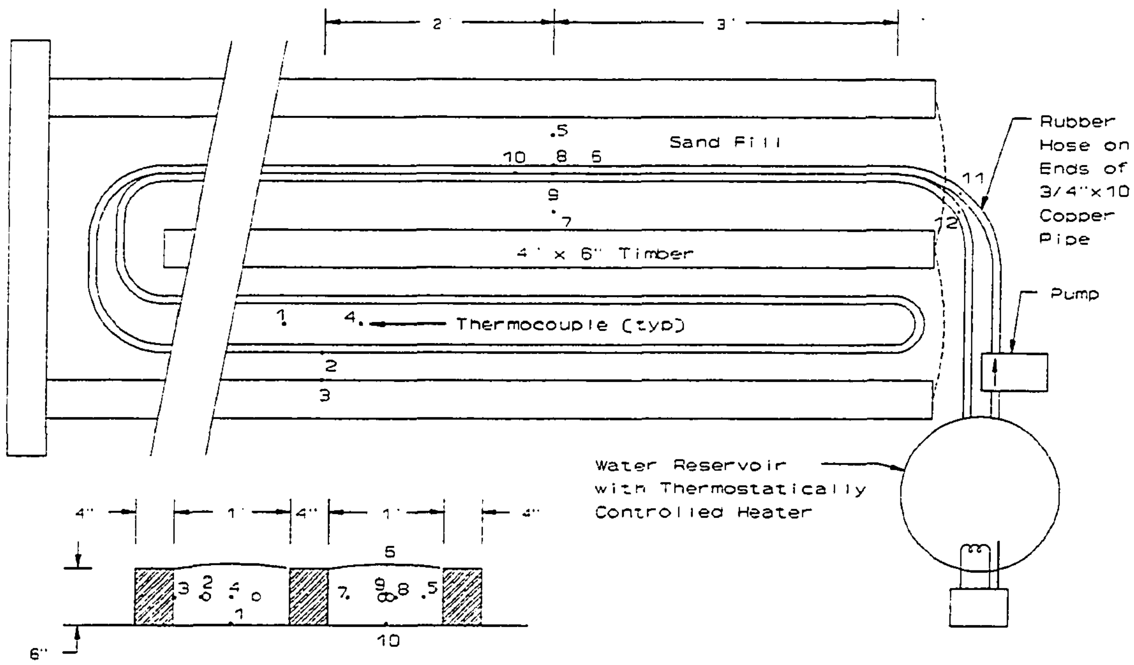


Figure 3.4: Layout for 1953 Cold Room Tests

that the heat loss from the pipes laid next to one another was equivalent to that from a single pipe with the same total cross sectional area. It was also apparently assumed that the heat loss from the rubber hose sections was negligible and that the system was at steady state with no freezing or thawing occurring in the sand. The computed thermal conductivities ranged from 0.5 to 2.2 Btu · in/hr · ft^2 · F° [0.04 to 0.18 Btu/hr · ft · F°; 0.07 to 0.32 W/m · C°] compared to a value of 1.9 Btu · in/hr · ft^2 · F° [0.16 Btu/hr · ft · F°; 0.27 W/m · C°] calculated using the average moisture content of 3.5 percent and a density for the sand of 71.2 lb$_m$/ft^3 [1,140 kg/m^3] using an emperical formula by Kersten (1949) for frozen, sandy soils. It is important to note that Kersten's data do not extend to the low moisture and density Page used and that Kersten estimated a degree of accuracy of $\pm$25 percent.

Page (1953) wrote that the computed conductivities seemed to increase with heat loss rate and noted that because "k is presumably a constant this indicates either the method of calculation or the testing technique is in error." Temperatures were only reported to the nearest 0.1 F°, and therefore the temperature differences may be assumed to be in error by as much as $\pm$0.2 F°. Because temperature differentials as small as 0.1 F° were used in calculations, the computed heat loss, and hence the computed thermal conductivity, may be off by a factor of three; no mention was made of this by either Page or Westfall.

According to Page, the tests were meant to determine the "minimum velocity required in the service pipes to prevent freezing" and temperatures of 33 to 34°F were used "because this was the range of temperatures anticipated in actual practice." Although the calculated heat transfer coefficients were not used in the design calculations, the tests did give the designers confidence that service lines could be kept from freezing and this led to a decision to run the pipes together to conserve heat and promote heat exchange between the supply and return lines. Page concluded:

> The minimum velocity of flow required through house service lines of copper or plastic to prevent freezing at soil temperatures below 32°F has not been satisfactorily established. It was concluded that velocities as low as 0.1 fps [0.15 gpm] had a good chance of successful operation at soil temperatures in the range of 25 to 27°F.
>
> The velocity of flow versus soil temperature relationship needs to be examined in more detail with the object of establishing the critical minimum velocity of flow for a series of temperatures from 0 to 32°F in various sizes of pipe and at different water temperatures.

3.2 DESIGN AND OPERATION OF THE FAIRBANKS SYSTEM

An interconnected network of mains was used rather than a single long, snaking loop or a number of separate loops. Such a network was far more complex, but it was cost effective, and it offered better fire flow protection, greater reliability, and flexibility of operation. The task of designing a network and ensuring minimum flow velocities in all segments would have been formidable had it not been for the recent development of an electrical analog method for analyzing piping networks. McIlroy, a professor of Electrical Engineering at Cornell University, designed a kit using a set of 200 different tungsten filament bulbs which he termed Fluistors. The voltage drop across any Fluistor varied as the 1.85 power of the current, giving an exact analogy to the Hazen-Williams formula for pipe flow. When the Fluistors were wired into a network to represent individual segments of pipe, a voltmeter or ammeter could be used to allow prediction of heads and flows. The effects of substitutions could also be quickly seen by noting the change in brightness of the bulbs. (Appleyard and Linaveaver, 1957; McIlroy, 1950 and 1951).

Two-speed pumps were selected for the Fairbanks water system to allow operation at either 2 or 3 fps; these velocities were believed to induce flows of 0.23 to 0.34 gpm in 200 feet of 0.79 inch ID service line. Uninsulated 6-inch diameter and larger wood stave pipe were used for the mains because it had some inherent insulating properties and reportedly could withstand freezing without damage. Copper service lines were used because no data were available on the toxicity of the plastic pipe being considered, and because copper lines could be thawed with resistance heating. The service lines were taped together to conserve heat, so an electric cable was attached to the lines at the main and run to the surface for thawing purposes. A 2-inch cover of vermiculite asphalt with a reported thermal conductivity of 0.67 Btu · in/hr · ft^2 · F° [0.056 Btu/hr · ft · F°; 0.10 W/m · C°] was used to insulate the service lines as shown in Figure 3.5a (Westfall and Wallace, 1953).

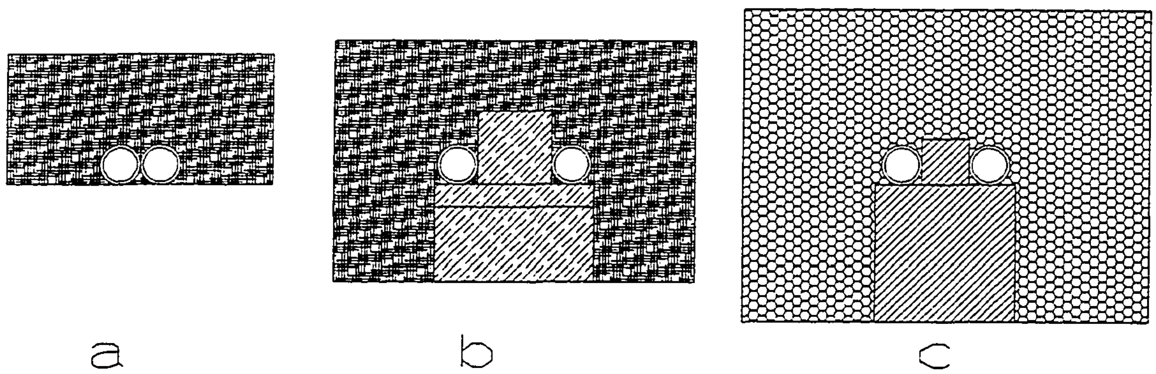


Figure 3.5: Fairbanks Service Line Insulation for ¾-inch Lines
a. 1953 design with vermiculite asphalt
b. 1955 design with spacers
c. 1991 design with urethane foam

The water treatment plant was located next to the electric power plant. Waste heat from the power plant was used to heat the water from 35°F up to a maximum of 58°F.

Pitorifices were installed horizontally a minimum of 8 inches apart. Service lines were sloped to drain to the main and the mains were sloped to drain to the fire hydrants. Fire hydrants were located directly on the mains with valves so that a section of main between two hydrants could be isolated and drained for maintenance while maintaining circulation with fire hoses connecting the hydrants. The fire hydrants did not have weep holes and had to be pumped dry and then filled with antifreeze.

The initial part of the system, consisting of five miles of wood stave water main, was put into operation in late December, 1953. Slow leaks occurred at many of the fire hydrant valves. The water did not mix with the antifreeze and nearly all the hydrants were removed and repaired during that first winter (Westfall, 1958). Air relief valves at the high points in the main also froze and were removed.

3.3 DESIGN CHANGES OVER THE PAST 40 YEARS

An additional 14 miles of main was installed in 1955; 12-gauge asphalt-dipped steel pipe insulated with a $^{9}/_{16}$-inch thickness of asphalt-impregnated asbestos felt was used as a cost-effective alternative to wood stave mains (Linck, 1961; Westfall, 1958). The service lines were separated from one another to allow elimination of the thaw connection cable, and the amount of insulation on the service lines was increased. The service lines were also supported at intervals with wood blocks to allow insulation to surround the lines as shown in Figure 3.5b.

In 1957, flow meters and thermocouples were installed in and along the mains, and attempts were made to determine flows in services for future design purposes (Page et al., 1957). In 1961, because the high speed pumping option was so seldom used, it was decided to design for 1.8 fps with a high speed option of 2.5 fps (Linck, 1961).

In 1968, new water mains were being field-foamed with blown foam insulation (Cornell, 1968). A 3-inch thick layer of field-foamed polyurethane with a thermal conductivity of 0.13 $Btu \cdot in/hr \cdot ft^2 \cdot F°$ [0.011 $Btu/hr \cdot ft \cdot F°$, 0.019 $W/m \cdot K°$] is currently specified for insulating services in Fairbanks as shown in Figure 2.5c. This insulation has performed well where the soil is dry; specimens up to 16 years old have shown negligible deterioration (Haigh, 1986).

In the early 1970's, a digital computer program became available for solving the complex hydraulic problems associated with the growing network of mains and in 1979, a computer model was developed for the city which was run on the Boeing Computer Service computer network. A new model was developed in 1989, for use on a personal computer (Roen, 1989).

The last of the 1953 wood stave pipe was replaced with ductile iron pipe (now standard) in the summer of 1991. While the wood stave mains had begun to leak excessively, the pitorifices and copper service lines showed no visible signs of wear or corrosion. Because the original pitorifices were designed to screw directly into the wood stave mains, they could not be salvaged, but the service lines were reconnected to new pitorifices installed in the new ductile iron main.

Pitorifices are currently installed in the side of the main angled 15° degrees up from the horizontal and about 6 inches apart. The upward angle allows more movement, lessening the effect of settlement. Due to problems with frost jacking, fire hydrants are now installed on tees off the main. When it is impractical to keep the distance short by jogging the main ciose to the hydrant location, a single upstream facing pitorifice diverts flow from the main through a 1-inch diameter copper line to the fire hydrant to ensure water circulation through the tee.

3.4 CORPS OF ENGINEERS STUDY

In 1977, the U.S. Army Corps of Engineers started a study for the redesign of a failed water system at Bethel, Alaska (Johnson, 1978). It was recognized that larger diameter pitorifices and service lines might allow use of longer service lines or lower flow rates in the mains. A 20-foot section of 6-inch diameter main was investigated both with no pitorifices or service line connections and with nominal ¾, 1, and 1¼-inch diameter pitorifices attached to service lines with a total looped length of 25 feet. Velocity in the main was 2.02 fps for eight trials and 4.20 fps for two trials. Flow rates through the main were measured with a 90-degree V-notch weir located downstream of the discharge throttling valve. Flow rates in the 25-foot line were determined by timing dye passage between two windows 24 feet apart. Water temperature was not recorded. A 0.9 fps flow in the 1-inch diameter service (assumed to be type L with ID=1.025 inches) was reported for a 2 fps flow in the main. The report concluded:

> Early bid advertising and contract award terminated the testing program before tests could be run with longer services and variable main velocities. ... The test data appended using 2.02 fps main velocity for service pipe velocities with increased lengths can be evaluated indicatively by using the Darcy-Weisbach formula. The Reynolds numbers are expected to be in the laminar, transition and turbulent flows so testing appears essential to validate results.

The U.S. Army Corps of Engineers study was the first to actually measure the pressure drop in the main due to the presence of pitorifices. This was done by measuring pressure drops with taps far upstream and downstream, both with and without pitorifices in the main. In the 1953 study, pressure drops were noted but the pressure tap locations were not specified and may have been close enough to the pitorifices to be influenced by their presence. See Appendix A for data from this study.

3.5 OTHER STUDIES

The only other significant research was done in 1965 and 1966 in Unalakleet (Murphy and Hartman, 1969; Ryan and Lauster, 1966) and during the summers of 1986, 1987 and 1988 in Anchorage, Alaska (Mauser, 1986 and 1987b; Harris, 1987; Spehalski, 1988). The Unalakleet study dealt mainly with heat losses, but did include the only accurate service line flow measurement made in the field up to that time. In Anchorage, Mauser used a 10-foot section of 4-inch diameter high density polyethylene main (ID=3.68 inches) and a 25-foot section of 1-inch diameter clear vinyl tubing with nominal 1-inch diameter pitorifices to obtain three data points in 1986. Flows in the service were estimated by timing the passage of injected dye. This proved to be very time consuming and inaccurate. The following year the vinyl tubing was replaced with a 100-foot length of nominal 1-inch diameter high density polyethylene service line (ID=1.076 inches) and velocities were estimated by taking timed samples and testing for conductivity after injecting an electrolyte. Finally, in 1988, both 100 and 200-foot lengths of service line were used and flows estimated by sensing the passage of an injected electrolyte past two isolated copper pieces of tubing. In all the tests, flow rates in the main were measured with a turbine type flow meter. Typical service line plumbing with angle stop valves and a tee was also used. Velocities in the main were varied from 1.2 to 2.3 fps resulting in service line velocities between 0.19 and 0.52 fps (0.54 to 1.47 gpm). Unfortunately, coiled lengths of tubing were used in these studies and the increase in head loss due to the coils was not taken into account. See Appendix A.

Crum (1977) estimated the head loss in the main due to the presence of pitorifices based on the measured total head loss for flows of 2.4 fps [0.72 m/s] in each of three looped 4-inch diameter mains in Kotzebue, Alaska. He then calculated the head loss expected from pipe and fittings and assumed the calculated difference to be due to the pitorifices pairs. This difference was an average of 0.11 ft water column (WC) per pitorifice pair. He suggested using 0.2 ft WC/pair as a conservative estimate for design purposes. Smith (1986) reported Crum's head loss values but Smith's velocities are 20 percent too high, apparently due to the use of British gallons instead of U.S. gallons when calculating velocities. It is interesting to compare this 0.2 ft WC/pair design value which was widely used, to 0.03 ft WC for nominal 1-inch diameter pitorifices in 6-inch diameter main at 2.0 fps which can be derived from experimental data from Johnson (1978) given in Appendix A.

3.6 THEORY OF OPERATION

Alter, Page, Westfall, and Johnson did not propose any theory for the operation of pitorifices beyond the assumptions already mentioned. Kanitz and Mace suggested some theoretical relations between service line and main velocities based on data from Page and Johnson (Mace, 1982). The first published theory was developed by Gerard and Smith (Smith, 1986 and 1991). They essentially noted that the head available can be expressed as:

$$H = K_{po}\frac{V_m^2}{2g} \tag{3.3}$$

where H is the head produced by the pitorifices, K_{po} is an experimentally determined value, and V_m is the velocity in the main. The head required for a given service line flow is:

$$H = \left(f\frac{L_s}{D_s} + \Sigma K_{ftgs}\right)\frac{V_s^2}{2g} \tag{3.4}$$

where H is the head required for a flow, f is the Darcy friction factor, L_s is the total service line length, D_s is the ID of the service line, K_{ftgs} are the fitting loss coefficients, and V_s is the flow velocity in the main. When the flow is laminar, the friction factor is given by:

$$f = \frac{64\,\nu}{V_s D_s} \tag{3.5}$$

where ν is the kinematic viscosity of water. Combining these equations and rearranging yields an expression for V_m:

$$V_m = \left(\frac{64\,\nu L_s}{V_s D_s^2} + \Sigma K_{ftgs}\right)^{0.5} \left(K_{po}\right)^{-0.5} V_s \tag{3.6}$$

A logical extension of their work is to write a quadratic expression and solve for V_s:

$$\left(\frac{\Sigma K_{ftgs}}{2}\right) V_s^2 + \left(\frac{32\,\nu L_s}{D_s^2}\right) V_s - \left(\frac{K_{po} V_m^2}{2}\right) = 0 \tag{3.7}$$

$$V_s = \left[\left(\frac{32\,\nu L_s}{D_s^2 \Sigma K_{ftgs}}\right)^2 + \frac{K_{po} V_m^2}{\Sigma K_{ftgs}}\right]^{0.5} - \frac{32\,\nu L_s}{D_s^2 \Sigma K_{ftgs}} \tag{3.8}$$

If the ΣK_{ftgs} term can be eliminated by using an equivalent length for fitting losses, Equation 3.6 can be written:

$$V_m = \frac{8}{D_s}\left(\frac{\nu\left(L_s + L_{eq}\right) V_s}{K_{po}}\right)^{0.5} \tag{3.9}$$

where L_{eq} is the equivalent length for fitting losses. Equation 3.9 can be solved for V_s:

$$V_s = \frac{K_{po} V_m^2 D_s^2}{64\,\nu\left(L_s + L_{eq}\right)} \tag{3.10}$$

Unfortunately, the available experimental data was insufficient to define K_{po}s for the range of pitorifice shapes and applications commonly used. Much of the service line flow data was taken under turbulent conditions with short service lines; with longer service lines flows would be laminar and total head loss could be expected to be greater. Also, values in the literature for K_{ftgs} and L_{eq} are usually given as constants and only apply to turbulent flow. These values do not apply to laminar flow conditions and are not constants with laminar flow.

3.7 CONCLUSIONS

The original tests on pitorifice performance were sufficient to prove the concept and allow the successful design of a unique system, but they were not sufficient to allow a good understanding of pitorifices. Very little systematic field testing was done in the following years and designs remained unchanged until the U.S. Public Health Service increased service line size from nominal ¾ to 1-inch diameter in the middle 1980s. However, they did not also drop flow rates and the rule of thumb remained 2 fps in the main and add a circulation pump at the house if line lengths exceeded 60 to 100 feet. Because thermal insulation has greatly improved, circulation pumps have become smaller and more efficient, and engineers generally know much more about heat transfer, it is important to reevaluate the pitorifice concept and gain a better understanding of its hydraulics. This will lead to better, more energy efficient designs.

CHAPTER 4: APPLYING THEORY TO PITORIFICE SYSTEM PERFORMANCE

A theoretical frame work is needed for designing experiments to determine pitorifice performance, evaluating alternative designs, and interpolating and extrapolating data. The performance of pitorifices can be modeled two ways: (i) they can be treated as fittings which have negative fitting or flow loss coefficients or (ii) they can be treated as a pump which has internal losses added to a head which is dependent on the velocity in the main.

Pitorifices are used in conjunction with a service line so it is also important to consider the service line as part of the system. The following sections discuss pitorifice and system hydraulics applied to modeling the complete system.

4.1 MODELING PITORIFICES AS FITTINGS

Hoopes et al. (1948), Idelchik (1986) and Vazsonyi (1944) present tables and figures giving loss coefficients for tee fittings as a function of geometry and the ratio of the side and combined channel flow rates. For some conditions the loss coefficients are negative indicating the addition of energy to one of the streams at the expense of the other. Elger et al. (1991) applied this approach in developing a new way to represent jet pump performance, noting that in "a generic sense, a wye or a tee in a piping network is a jet pump. This is because a wye or a tee combines a high and low speed fluid flow to produce a mixed fluid flow leaving a device."

This approach could be applied to the pitorifice pair. A function for the combined gain coefficient could be determined for a given pitorifice shape:

$$K_{gain} = F_1\left(V_m,\ V_s,\ \mu,\ \rho,\ D_m,\ D_{po},\ x_s,\ x_i\right) \tag{4.1}$$

where V_m is the velocity in the main, V_s is the velocity in the service line, μ is the absolute viscosity, ρ is the density, D_m is the inside diameter of the main, D_{po} is the outside diameter of the pitorifices, x_s is the separation distance between pitorifices, and x_i is the insertion depth of the pitorifices into the main. This function could then be equated to a function for the combined service line and fittings loss coefficient:

$$K_{loss} = F_2\left(V_s,\ L_s,\ D_s,\ fittings\right) = f\frac{L_s}{D_s} + \Sigma K_{ftgs} = f\frac{\left(L_s + L_{eq}\right)}{D_s} \tag{4.2}$$

where L_s is the length of service line, D_s is the service line inside diameter, fittings refers to the elbows, tees, unions, etcetera (excluding the pitorifices) in the service line, f is the Darcy friction factor for the service line, ΣK_{ftgs} is the sum of fitting loss coefficients in the service line, and L_{eq} is the equivalent length of all the fittings in the service line.

Setting the two functions equal to one another would allow determination of V_s given values for all the other variables. However, because function F_1 is empirical it might be better to represent both functions by families of curves as a function of V_s. An alternative to graphing K_{gain} and K_{loss} is to multiply each by $V_s^2/2g$ to obtain a performance curve (analogous to a pump curve) and a system curve respectively.

Both functions can also be described in terms of dimensionless numbers. Application of the Buckingham pi theorem to Equation 4.1 leads to the prediction that five dimensionless groups would serve to define K_{gain}. If density is considered constant and the separation distance assumed to have negligible effect, three dimensionless groups can be used. The parameters for performance and systems curves can also be expressed as dimensionless numbers.

4.2 MODELING PITORIFICES AS A PUMP WITH INTERNAL LOSSES

A pump may be modeled in a manner analogous to that of an electric battery with internal resistance in an electric circuit by considering the pump as a constant head device in series with an internal hydraulic resistance. The pitorifice pair can therefore be modeled as a combination of a pump with a flat curve from which is subtracted a system curve representing losses in performance due to the flow through the pitorifices and to internal losses in the pitorifices. If these losses are then added to the actual system curve for the service line (given by $F_2V_s^2/2g$) only the shut-off head for the particular shape of the pitorifice pair needs to be known as a function of V_m for different geometries. This greatly simplifies the analysis and presentation of performance data. The performance of a given shape can then be adequately described in terms of as few as three dimensionless variables.

Using an analytical approach, the shut-off head can be considered as the sum of three different heads. The upstream pitorifice (uppit) can be treated as a type of pitot tube which contributes velocity head. The downstream pitorifice (downpit) can be treated as a device which exploits both the pressure drop in its turbulent wake caused by pressure drag and the orifice or venturi effect due to the flow constriction caused by its presence. Finally, the head loss due to frictional drag can then be considered for the pair of pitorifices combined. The following sections deal with these individually and with losses in the service line.

4.3 HEAD FROM UPPIT

Just about any upstream facing opening in a reasonable geometrical shape will provide a head close to the velocity head (Folsom, 1956). The velocity head, H, is given by:

$$H = \frac{V^2}{2g} \tag{4.3}$$

where V is the velocity of the fluid at the opening and g is the acceleration of gravity.

A correction factor, K, can be used as a coefficient with V^2. For a pitot tube in an unconfined flow, K can be slightly larger or smaller than 1, and is a function of Reynolds number, turbulence, orientation, and shape of the pitot tube (Goldstein, 1965). In an ideal pitot tube the opening is upstream of the stem to reduce the effects of the stem, but the uppit's opening is right on the stem itself. For an uppit, K might be greater than 1 if the venturi effect is significant or less than 1 due to unfavorable shape.

In a straight section of pipe with fully developed flow, the maximum velocity occurs in the center and decreases to zero at the wall. The velocity at a given radius, V_r, in terms of the fractional distance from the center relative to the distance to the wall, r/R, and the average velocity in the main, V_m, for two-dimensional stabilized turbulent flow can be approximated by (Idelchik, 1986):

$$V_r = k\left[1 - \frac{r}{R}\right]^{\frac{1}{n}} V_m \tag{4.4}$$

$$\begin{aligned} \textit{where}\quad & k = 1.26,\ n = 6\ \textit{for}\ Re = 4{,}000 \\ & k = 1.23,\ n = 7\ \textit{for}\ Re = 25{,}000 \\ & k = 1.20,\ n = 8\ \textit{for}\ Re = 200{,}000 \end{aligned}$$

Because the velocity at the uppit entrance varies with its position, if V_m is used in determining velocity head a correction needs to be applied. Another correction must be applied to account for blocking of the flow, wall effects, and velocity and pressure gradients. For purposes of this analysis, it is assumed that all the adjustments discussed above to Equation 4.3 can be incorporated into the single correction factor, K_{up}, to give a velocity head at the uppit:

$$H_{up} = K_{up}\frac{V_m^2}{2g} \tag{4.5}$$

4.4 HEAD FROM VENTURI EFFECT

A venturi effect may be significant (Folsom, 1956). The venturi effect refers to the drop in static head from an increase in velocity head due to an obstruction in the main. The static head due to this effect can be calculated using:

$$H_v = \frac{\alpha_{dn} V_{dn}^2 - \alpha_m V_m^2}{2g} \tag{4.6}$$

where H_v is the depression in the static head due to the venturi effect, α is the kinetic-energy correction factor, V is the average velocity in the section, and the subscripts dn and m refer to locations in the main at the downpit and at an unobstructed cross-section. The kinetic-energy correction factor is given by:

$$\alpha = \frac{1}{A} \int \left(\frac{V_r}{V_m} \right)^3 dA \tag{4.7}$$

where A is the area of the section. Using Equation 4.4 for Re = 200,000 yields $\alpha = 1.06$.

Using the continuity equation for steady, incompressible flow:

$$V_{dn} = \left(\frac{A_m}{A_{dn}} \right) V_m \tag{4.8}$$

defining a venturi coefficient as:

$$K_v = \frac{A_m^2}{A_{dn}^2} - 1 \tag{4.9}$$

and assuming $\alpha_{dn} = \alpha_m = 1$, Equation 4.6 can be written:

$$H_v = K_v \frac{V_m^2}{2g} \tag{4.10}$$

It is important to note that this drop in head assumes steady, frictionless flow and no Vena contracta and ignores the effect of losses and disturbances due to friction and turbulence which are normally accounted for in formulas for orifice or venturi performance. The drop would also have to be measured perpendicular to the flow and not on the curved surface of the downpit. However, the analysis may offer insight into the magnitude of the pressure drop. Calculated values of K_v are given in Table 4.1. For a 0.75-inch OD pitorifice inserted two inches into a 6 or 8-inch diameter main, which is common practice in Fairbanks, the effect is small. For a 0.9375-inch OD pitorifice in a 4-inch diameter main, which is fairly standard for new construction in the villages, the effect is over three times greater and close to one third that presumed to be obtained from the uppit.

Table 4.1: Calculated K_vs for Pitorifices in Mains

Type K Tube Size	Pitorifice Size		Outer Bend Radius	K_v for 2" Projection into Main with ID=			
	ID	OD		4.0"	6.0"	8.0"	10.0"
⅝"	0.6520"	0.7500"	2"	0.25	0.10	0.06	0.04
	0.6965"	0.8125"	2"	0.27	0.11	0.06	0.04
¾"	0.7450"	0.8750"	2.25"	0.30	0.12	0.06	0.04
	0.8135"	0.9375"	2.25"	0.32	0.13	0.07	0.04
1"	0.9950"	1.1250"	2.25"	0.39	0.15	0.08	0.05
1¼"	1.2450"	1.3750"	2.50"	0.47	0.18	0.10	0.06
1½"	1.4810"	1.6250"	2.75"	0.56	0.21	0.11	0.07
2"	1.9590"	2.1250"	3.25"	0.68	0.26	0.14	0.09

4.5 HEAD FROM DOWNPIT

Pressure drag refers to the depression in pressure at the downpit due to the turbulent wake area downstream where there is boundary layer separation in the flow. The pressure distribution can be described in terms of a dimensionless surface-pressure coefficient:

$$C_p = \frac{P - P_\infty}{\frac{1}{2}\rho V_\infty^2} \tag{4.11}$$

where P is the pressure at the surface, P_∞ is the free stream pressure, V_∞ is the free stream velocity, and ρ is the fluid density. Pressure is related to head by $P=g\rho H$ so a head from pressure drag can be expressed as:

$$H_p = \frac{C_p V_\infty^2}{2g} \tag{4.12}$$

Figure 4.1 shows C_p as a function of location and Reynolds number for unconfined flows past a cylinder. The Reynolds number is about 10,000 for 2-fps flow past a 1-inch diameter tube so C_p should be about -1 for a pitorifice. The theoretical surface-pressure coefficient is given by $C_p = 1 - 4\sin^2\theta$. This assumes no flow separation and is valid only for inviscid flow. The experimental values are time-averaged because the pressure was unsteady and pulsating (White, 1974). The boundary layer for a circular cylinder becomes turbulent at Re ≈ 350,000 (Daugherty et al., 1985). The subcritical and supercritical designations refer to regions below and above Re ≈ 350,000.

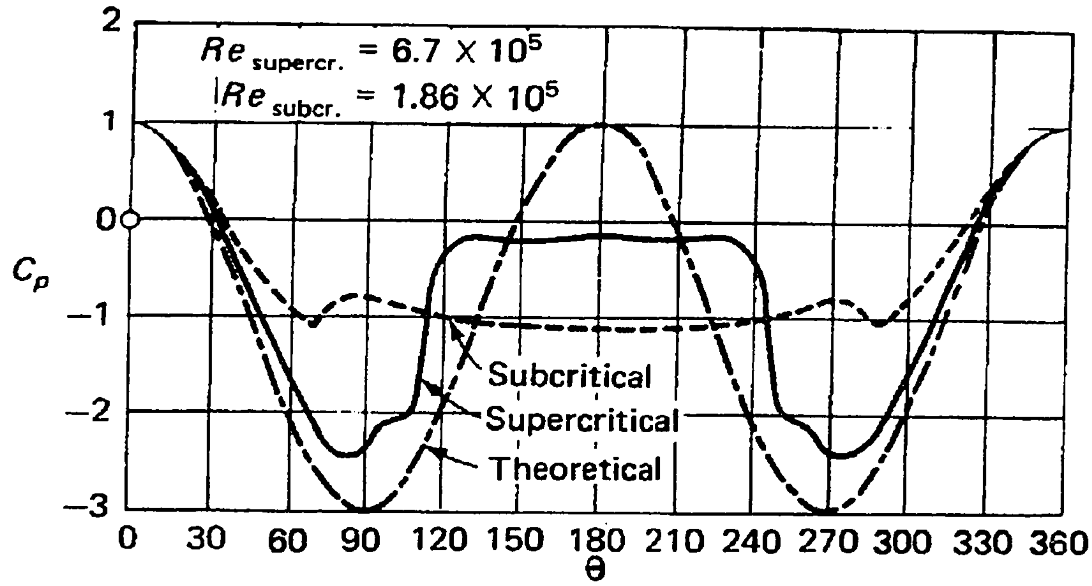


Figure 4.1: Pressure Distribution on a Cylinder in Unconfined Flow (White, 1974)

Cole (1935) developed the insertion flowmeter with identical upstream and downstream facing pitot elements shown in Figure 4.2. He reported the trailing orifice was about one-half as efficient as the upstream orifice. This implies the surface-pressure coefficient was −0.5 for this shape. It should be kept in mind that the insertion flow meter has openings that point directly up and down-stream, whereas a pitorifice opening can extend over a 120° arc.

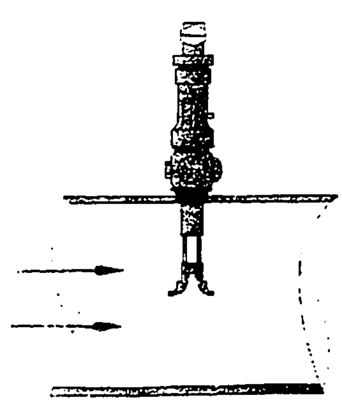


Figure 4.2: Bi-directional Insertion Pitot Meter (Cole, 1935)

Although an insertion flowmeter is in a confined flow, it is small relative to the pipe so the effects of confinement can be expected to be limited. Such may not be the case for transverse pitot tubes or averaging pitot meters which consist of tubes which span the pipe diameter as shown in Figure 4.3.

Figure 4.4 is from a study on transverse pitot tubes used for pipe flow measurements (Christiansen, 1937) and Table 4.2 was derived using data from the same study.

C_p may be considered a function of K_v. Cole (1935) determined that a correction amounting to one-third of the area of the projected rod was needed to account for probe blockage. This suggests that a value for C_p for the confined case may be a function of C_p for the unconfined case and K_v. The K_vs calculated for cases (a) and (c) in Figure 4.6 and Table 4.2 most closely compare with K_vs for pitorifices. The shape of the projection and the influence of surrounding surfaces can cause the surface pressure distribution to vary greatly, so a pitorifice in a pipe can not be expected to behave in the same way as even a confined cylinder. All or only part of the projected area may be needed to calculate a suitable K_v.

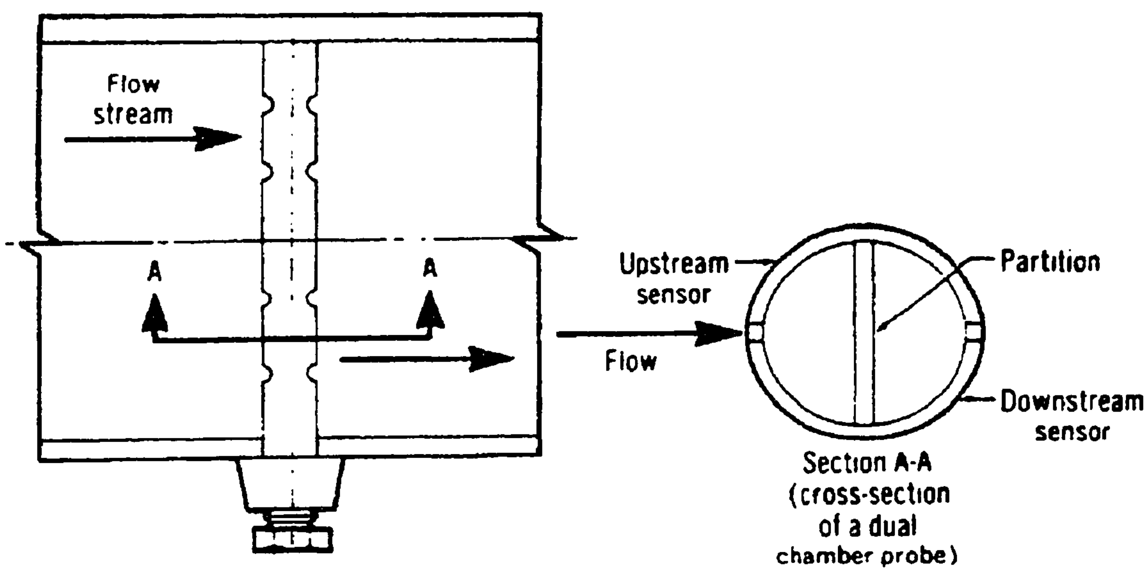


Figure 4.3: Bi-directional Averaging Pitot Meter (Bean, 1971)

Table 4.2: Values Derived from Study by Christiansen (1937)

Case	Diameters: Tube in.	Pipe in.	dP at 0° in. WC	V fps	Re for tube	dP at 180° in. WC	C_p at 180°	K_v
a)	0.3125	5.625	13.9	8.64	18500	-15.2	-1.09	0.16
b)	0.3125	1.04	7.5	6.34	13600	-12.6	-1.68	1.70
c)	0.125	1.04	5.7	5.53	4700	-4.4	-0.77	0.41

Note: V was calculated by assuming K_{up}=1

4.6 HEAD FROM LOSSES IN MAIN

Frictional drag head losses will depend on geometry and Reynolds number. They are difficult to estimate and should be experimentally determined. The frictional drag coefficient can be defined as:

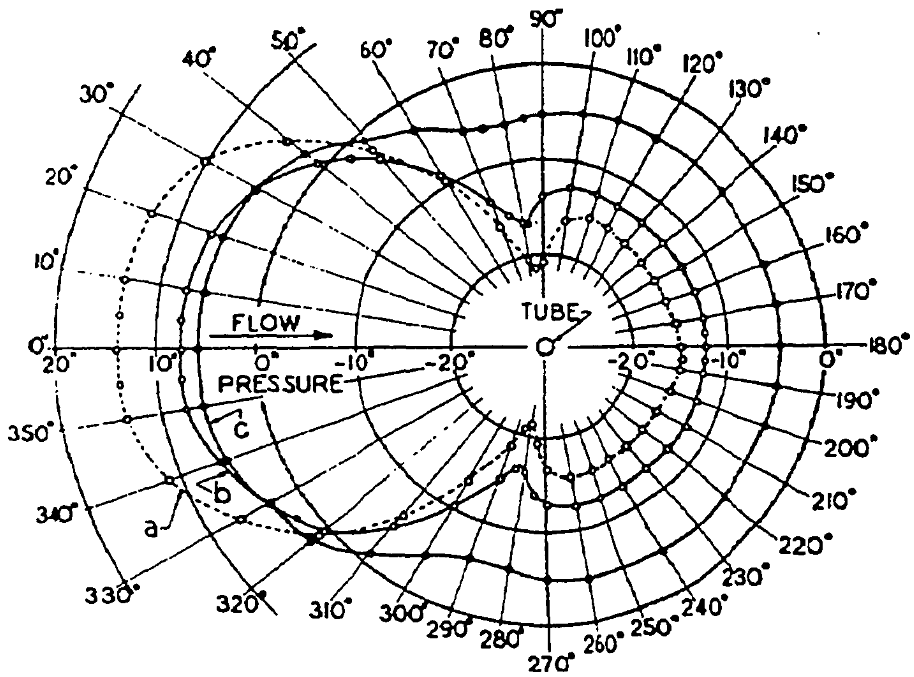


Figure 4.4: Pressure Distribution on Transverse Tubes in Pipes (Christiansen, 1937)

$$C_f = \frac{\Delta P}{\frac{1}{2}\rho V^2} \tag{4.13}$$

Because some of the drag may be developed downstream of the pitorifice pair, all the head loss may not be utilized by the pitorifices. There will also be a small amount of head loss in the main due to the small distance between the pitorifices.

4.7 TOTAL HEAD FROM PITORIFICES

From the preceding discussions on the various factors that could contribute to head difference across the pitorifice pair it is apparent that the total head can be related to a single coefficient for the velocity head in the main which in turn would be a function of geometry and Reynolds number:

$$H_{po} = K_{po}\frac{V_m^2}{2g} \tag{4.14}$$

where the subscript po refers to the pitorifice pair. The losses due to flow in the pitorifices must be subtracted from this head. These losses can be incorporated into K_{po} making it a function of V_s also as discussed in Section 4.1, but it is more straightforward to consider these losses as service line losses. Assuming flow and performance losses in the pitorifices can be treated as fitting losses, K_{po} may be written as:

$$K_{po} = K_{up} - C_p + C_f^{*} \tag{4.15}$$

where all the terms are experimentally determined and C_f^{*} represents only that part of C_f which is utilized by the pitorifices. All the terms may depend on the insertion depth, the relative size of the pitorifices and the pipe, the shape of the pitorifices, and the spacing and orientation as well as on the viscosity of the water.

4.8 TOTAL HEAD REQUIRED BY SERVICE LINE

The service line losses are presumed to include frictional losses in the pipe itself including losses due to bends in the pipe and losses due to fittings. The pitorifices are considered as fittings in the service line as discussed in Sections 4.2 and 4.7. The following sections deal with service line losses.

4.9 HEAD LOSS IN SERVICE LINE

The Darcy-Weisbach equation for head loss in a pipe provides the definition of the Darcy friction factor, f:

$$H = f\frac{L}{d}\frac{V^2}{2g} \tag{4.16}$$

For stabilized laminar flow (Re<2,000) the friction factor is given by Poiseuille's Law:

$$f = \frac{64}{Re} \tag{4.17}$$

and for turbulent flow (Re>4,000) it can be approximated by (Haaland, 1983):

$$f = \left(1.8\log\left[\frac{6.9}{Re} + \left(\frac{\epsilon/d}{3.7}\right)^{1.11}\right]\right)^{-2} \tag{4.18}$$

where ϵ is an experimentally measured value for hydraulic roughness. For commercial drawn tubing ϵ=0.000,005 feet [0.0014 mm] (Moody, 1944).

For flows with 2,000<Re<4,000 the friction factor increases from that given for laminar flow to that given for turbulent flow. There is no equation given in the literature for this region although data have been reported by various investigators. Hinze (1975) reported on experiments that suggest transition to turbulence is not a sudden phenomenon and that the critical Reynolds number ranges over $2{,}000 < Re_{crit} < 2{,}400$.

If a constant head is available between two ends of a pipe, as is the case with pitorifice service lines, it is possible that the flow rate will oscillate as turbulent plugs are formed and passed out of the pipe when flows rates are in the critical zone. This happens because, as the laminar flow rate increases, the critical point is passed and turbulence is initiated. This in turn results in higher resistance and a drop in flow. As the flow rate drops below the critical point, it becomes laminar again, and the cycle repeats.

If the water enters the uppit turbulent or if the entrance isn't smooth, turbulence can occur for a distance at low Reynolds numbers. Moody (1944) reported that when "there is distinct turbulence in the entering fluid, the flow in the critical zone is likely to be pulsating (Prandtl and Tietjens, 1934) rather than steady. The effects of strong initial turbulence may even extend into the laminar-flow zone, raising the f values somewhat, as far as to a Reynolds number of about 1,200."

The use of a sharp entrance or the omission of a calming length caused some problems with early researchers. Drew et al. (1932) reported that the higher values of f shown in curve D of Figure 4.5 reported by some workers, resulted from the use of too short a calming length. He went on to say:

> With a given apparatus operated under a given set of conditions there is found a more or less definite value of the Reynolds number above which wide deviations from [Poiseuille's Law] occur. This value, long ago identified by Reynolds as the point where normal stream-line flow ceases, is the Reynolds criterion, Re_C, for the apparatus under the given conditions. It varies with the apparatus and method of testing. When f's, determined for a given set of circumstances, are plotted as ordinates against Re as abscissae, the points for $Re > Re_C$ lie at first on a steeply rising curve, such as in the curve B in [Figure 4.5], until the gently dropping curve C typical of turbulent flow is reached.
>
> That steady turbulent flow is impossible in any round pipe below a definite value, Re_{LC}, of the Reynolds number may be regarded as established. In other words, the Reynolds criterion of a particular apparatus cannot be less than Re_{LC}. Schiller [e.g. Schiller, 1920] and his co-workers have studied this matter extensively, and in their own experiments have been unable to induce turbulent flow below Re = 2320. To be consistent with the foreign literature the name "Lower Reynolds Criterion" will be given to this limiting Reynolds number. The value of $Re_{LC} = 2100$ is adopted for the present.

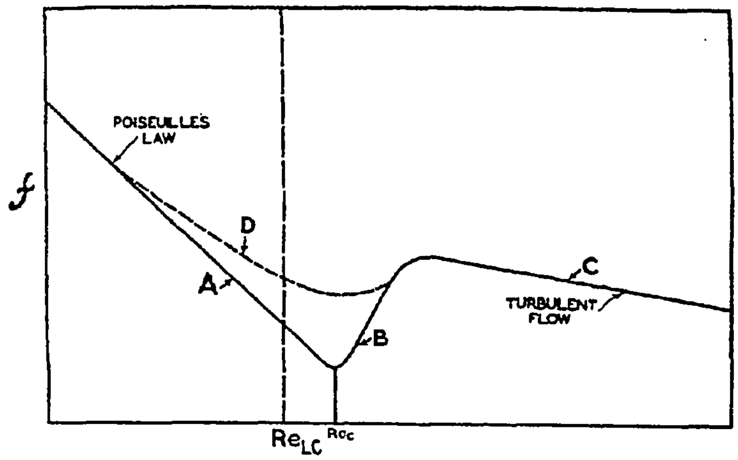


Figure 4.5: Typical Friction Factor Curve (Drew, 1932)

Based on these observations it seems reasonable to assume that for Re < 1,200 Poiseuille's Law can be used but for Re > 1,200 there may be oscillations and turbulence that would increase the effective value of f.

Although previous investigators focussed on service line velocities, it is mass or volumetric flow rates that are important. Assuming 2 Btu/hr · ft heat loss and a 1 F° temperature drop permitted in a service line, the required mass flow rate for a 50-foot main to house run will be 100 lbs/hr or 0.2 gpm. This results in flow velocities well within the laminar region for common service line sizes. For nominal ¾ and 1-inch diameter copper water tube, for example, the Reynolds numbers are 650 and 490 respectively at 50°F, the highest temperature normally used, and lower at lower temperatures. The required head can be calculated by noting:

$$V_s = \frac{\dot{m}_s}{\rho} \frac{4}{\pi D_s^4} \tag{4.19}$$

where $\dot{m}_s$ is the service line mass flow rate. Substituting Equations 3.5, and 4.19 into Equation 4.16 gives:

$$H = \frac{128\,\nu}{g\rho\pi} \frac{L_s \dot{m}_s}{D_s^4} \tag{4.20}$$

where ν is the kinematic viscosity of water. For a constant length and mass flow rate:

$$\frac{H_1}{H_2} = \left(\frac{D_{s2}}{D_{s1}}\right)^4 \tag{4.21}$$

where the subscripts 1 and 2 refer to two different sizes of line. A 1-inch diameter service line would therefore require only 0.3 the head of a ¾-inch diameter service line. Because the head available from pitorifices varies roughly with the square of the velocity in the main, the velocity in the main required for the 1-inch diameter service would be 0.56 of that needed for the ¾-inch diameter service. This would in turn result in about one seventh the energy use for pumping in the main.

4.10 HEAD LOSSES IN BENDS

When water flows through a bend as shown in Figure 4.6, swirling secondary flow due to centripetal acceleration and possible flow separation at the walls result in additional frictional loss.

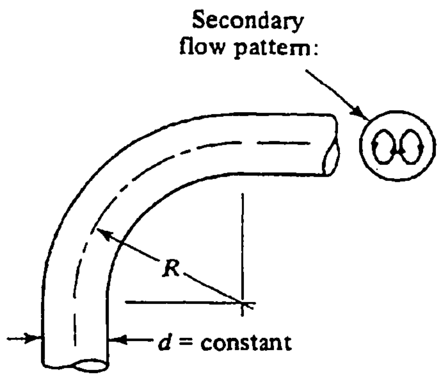


Figure 4.6: Flow Pattern in a Bend (White, 1986)

Idelchik (1986) uses λ_{el}, which is equivalent in meaning to f, for computing the total resistance in bends. His formulas are valid for R/d≥3 and he presents graphed solutions 3<R/d<50 and 400<Re<100,000 so the formulas are presumably valid over that range, although the data they are based on may not cover the extremes for all combinations of R/d and Re. Idelchik gives the following formulas for determining λ_{el}:

$$\begin{aligned} \lambda_{el} &= \frac{20}{Re^{0.65}}\left(\frac{d}{2R}\right)^{0.175} && \textit{for} \quad 50 < Re\sqrt{\frac{d}{2R}} < 600 \\ &= \frac{10.4}{Re^{0.55}}\left(\frac{d}{2R}\right)^{0.225} && \textit{for} \quad 600 < Re\sqrt{\frac{d}{2R}} < 1400 \\ &= \frac{5}{Re^{0.45}}\left(\frac{d}{2R}\right)^{0.275} && \textit{for} \quad 1400 < Re\sqrt{\frac{d}{2R}} < 5000 \end{aligned} \qquad (4.22)$$

The head loss in a 90° bend in 0.745-inch ID pipe with R/d=5 for a length of 0.5 feet (calculated as the arc length of the bend along the axis) will be two to three times that of the head loss in the same length of straight pipe in laminar flow. Adding several feet of equivalent length should account for loss from bends.

4.11 HEAD LOSSES IN FITTINGS

The engineer can use resistance coefficients, equivalent lengths, or flow coefficients to arrive at head losses due to fittings. The use of flow coefficients is usually confined to valves and will not be considered here. Unfortunately, most of the values reported in the literature are not valid for laminar flows, having been obtained under turbulent flow conditions. The fitting resistance coefficient, K_{ftg}, is defined as:

$$K_{ftg} = \frac{H_{ftg}}{V_{ftg}^2/2g} \tag{4.23}$$

where H_{ftg} is the additional head loss which occurs when a fitting is substituted for a section of straight pipe in a run and V_{ftg} is the velocity in the fitting.

As early as 1892, extensive experiments were performed which showed a variation in K_{ftg} with Reynolds number (Freeman, 1941). However, due to the wide variety of fittings and the effect of installation which makes estimated losses uncertain, this variation is usually not taken into account. Streeter (1947) noted:

> A given fitting may have many different energy losses for a given flow of a specified fluid, depending upon the upstream and downstream conduit arrangements. As there is no limit to all these possibilities, the losses cannot be predicted closely unless a test has been run on a similar system. In general, fitting losses are determined with straight conduit both upstream and downstream for at least 30 diameters. The experimental results from these tests give little information as to the loss caused by the fitting when it is preceded or followed closely by other fittings.

The resistance coefficient is usually not correlated in the literature with Reynolds Number or roughness, and data are usually for turbulent flow conditions (White, 1986). References commonly consulted by engineers (ASHRAE, 1989; Hicks and Hicks, 1985; Hicks and Edwards, 1986; Hydraulic Institute, 1961; Merritt, 1976; Nayyar, 1992; Williams and Culp, 1986) list values that vary by 50 percent or more from each other. The Crane Co.'s Technical Paper No. 410 (Crane, 1988) is often quoted as a source of information on loss coefficients. They advise the "resistance coefficient K is ... considered as being independent of friction factor or Reynolds number, and may be treated as a constant for any given obstruction (i.e., valve or fitting) in a piping system under all conditions of flow, including laminar flow."

Crane gives most of the resistance coefficients as multiples of the friction factor for flow in the zone of complete turbulence (f_T) for a pipe with the same diameter as the fitting. This allows them to give one value for K_{ftg} in terms of an equivalent L/D and f_T for a type of fitting whereas others report K_{ftg}s for each size of fitting. Crane supports this approach by giving a graph of log(size) as a function of log(K_{ftg}) or log(f_T). The slopes of the lines for the various types of fittings are in good agreement with the slope of the line for pipe alone. This is reasonable because the relative roughness of pipe and fittings for a given size and material will probably vary the same way.

It is reasonable to assume that even if the flow in the pipe in a given situation is not in the zone of complete turbulence, the flow in the fitting may be considered completely turbulent. Unfortunately, this approach is not valid for hydraulically smooth pipe. It was also noted by Pigott (1950) that threaded fittings may have less loss than flanged or sweat fittings because their larger ID can more than offset the losses due to sudden expansion and contraction. In fact the head loss at low Reynolds numbers may be primarily due to expansion and contraction. Idelchik (1986) gives graphs for losses due to sudden expansion and contraction which show a strong dependence on Reynolds number. This suggests that Crane (1988) is wrong when they advise the use of the same resistance coefficient for laminar that is used for turbulent flow.

Fitting losses for service lines will have to be experimentally determined with the expectation that they will vary with the Reynolds number. Idelchik (1986) suggests that for the general case K_{ftg} be expressed as the sum of two terms:

$$K_{ftg} = \frac{K_1}{Re} + K_2 \tag{4.24}$$

At very low Reynolds numbers the first term dominates, and at very high Reynolds numbers it can be neglected. This effect is quite clear in the work of Jamison and Villemonte (1971).

An alternative approach would be to use equivalent lengths for fittings. Constance (1979) published a table giving equivalent lengths for threaded fittings starting at 1½-inch but he stated the values apply only to turbulent flow. Nibco Inc. (1976) publishes a table for wrought-copper and cast-brass fittings from ⅜ to 6-inch in their catalog. The information is based on tests conducted at Harvard University in 1948 but they did not mention any limitations nor did they give a more complete citation. White (1986) declined to pursue the concept of equivalent length calling it "an artificial concept."

4.12 CONCLUSIONS

Using different designs for the uppit and downpit would improve performance. The greatest potential improvement would be to have two openings in the downpit oriented perpendicular to the flow in the main instead of a single opening facing downstream. However, having two different shapes for pitorifices would increase the likelihood of installation errors as well as increase inventory requirements. The potential does exist, however, that a single shape can be found which will perform better than the shapes commonly used. If a different shape costs more to produce, this additional cost must be offset by reduced operating costs.

Increasing the projected area in the main by increasing size or insertion depth of pitorifices will also improve performance. This will, however, be at the expense of increased head loss in the main. It is possible that the effect of reduced flow requirements in the main will more than offset any local increase in head loss, but experimental data are needed to prove this.

Increasing service line diameter would provide the greatest improvement to system performance. Equation 4.21 shows that nominal 1-inch service lines would require only 30 percent the head which ¾-inch lines require. Assuming the head available is directly related to the square of the velocity in the main, it can be shown that savings in energy costs for pumping will offset the increased capital cost of the larger service lines in most installations.

CHAPTER 5: TEST EQUIPMENT AND PROCEDURES

The following sections describe the selection and use of various components of test equipment and the construction and use of apparatus built for the study.

5.1 PRESSURE MEASUREMENT

Instruments for measuring differential pressures from 0 to 75 mm [0 to 3 inches] of water column (WC) and 0 to 3 m [0 to 10 ft] WC were needed for pitorifice and small pump performance testing respectively. Instruments able to withstand water system line pressures as high as 1,000 kPa [150 psi] were required for field testing applications. Finally, line pressure recording instruments were desired. Accuracies better than ± 5 percent, continuous readout, and 4 to 20 mA or 0 to 5 volt outputs were required in most cases.

A literature search was conducted because difficulties with measuring small pressure differentials in water were anticipated. Brombacher (1970), Demorest (1985), Lior (1986), and Slomiana (1979) gave reviews of various means of measuring pressure differentials but did not specifically address the problem of measuring very small pressure differences in water. Lin (1976) described the construction of a dual fluid manometer but ignored the effects of changing surface conditions. Nakagawa (1980) photographed tracer particles in precision bore glass tube under laminar flow conditions to allow determination of flow velocity and calculation of pressure drop. Preston (1972) mentioned the disadvantages in using a micromanometer with a heavier than water liquid such as carbon tetrachloride or ethylene dibromide and proposed a device with two trapped air reservoirs in conjunction with a standard manometer designed for air flow for use with water. Weihs and Sumer (1973) suggested modifications to Preston's apparatus.

Problems with pressure fluctuations due to vortex shedding from the pitorifices and pump vibration were also expected. Falkner (1935) reported irregular movement in the buoyant float of the Bentzel velocity tube when used to measure turbulently flowing water and previous experiments on pitorifices showed manometers to be difficult to read due to fluctuating levels. Cole (1935) and Walski (1984) reported problems measuring differential pressures in water mains due to pressure fluctuations. Vortex shedding from a circular cylinder has a Strouhal number ($St = fd/V$) of about 0.2 for Reynolds numbers between 100 and 100,000 (White, 1986). This yields a frequency on the order of 10 Hz for flows past a pitorifice for typical sizes.

5.1.1 Manometers

One of the main accuracy limitations of a manometer is measuring the difference in menisci levels. Accuracy can be increased by using a point or hook gage on a micrometer, by inclining the manometer, or by using a second fluid with a density close to that of the process fluid. Figure 5.1 shows standard and inverted manometers, a micromanometer, and an inclined manometer. While all manometers have two fluids, a dual-fluid manometer deliberately exploits density differences between fluids to obtain better precision in measurement. The dual-fluid manometer can be a standard U-type with a fluid denser than the process fluid trapped in the bottom, or it can be an inverted U-type with a less dense fluid trapped in the top.

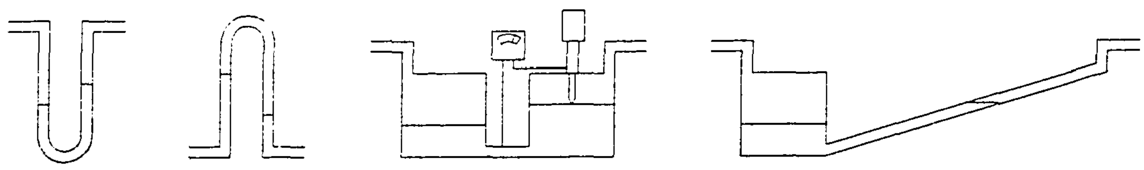


Figure 5.1: Major Types of Manometers

Surface tension and fluid to solid contact angles can cause problems with reading any manometer. The height a fluid will rise in a tube is approximated by:

$$h = \frac{2\Upsilon\cos\theta}{\rho g R} \tag{5.1}$$

where h is the height, Υ is the surface tension, θ is the fluid to solid contact angle, ρ is the fluid density, g is the acceleration of gravity, and R is the inside radius of the tube. The easiest way to minimize these effects is to increase R. Micromanometers use precision bores of large diameter, a probe attached to a micrometer, and a continuity indicator. The trapped manometer fluid is normally treated to reduce surface tension and increase electrical conductivity and therefore the micromanometer is more appropriately used where the process fluid is a gas.

Corrections due to temperature changes can be made relatively easily if the thermal expansion coefficients of the fluid and the container are known. Changes in system pressure can affect the volume of the trapped fluid containment, particularly with flexible plastic lines and higher pressures, and this may also have to be taken into account. An inclined tube can be used to increase the distance measured for a given pressure drop but the surface area contacted by the fluid is also increased and any problems with surface effects and contamination are magnified.

A Microtector model micromanometer manufactured by Dwyer Instruments, Inc. (Michigan City, Indiana) was available from the University of Alaska Fairbanks so it was used for calibrating dual-fluid manometers, differential pressure gauges, and differential pressure transducers. The Microtector is accurate and repeatable to within ± 0.02 mm of WC throughout its 0 to 50-mm WC range. It utilizes a micrometer probe to make contact with water containing a dye additive and contact is indicated with a micro-ammeter.

Calibration was done by pressurizing air slightly above atmospheric pressure by adjusting the water level in a U-tube as shown in Figure 5.2. Water was added to the open end to trap and pressurize air in the vinyl tubing connecting the micromanometer to the instrument being calibrated. An air bleed valve and a drain valve were used to adjust pressure.

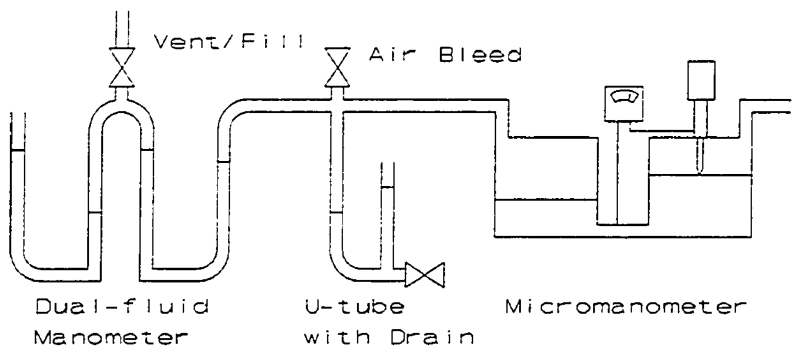


Figure 5.2: Use of Micromanometer for Calibration of Dual-fluid Manometer

While there are many manufacturers for manometers, all models identified which would measure small pressure differences were for use with gases. Preston (1972) and Weihs and Sumer (1973) added air reservoirs to air-type inclined manometers to measure very low velocity heads in water with a pitot tube, but, while very accurate, these had slow response times and were sensitive to temperature changes. The possibility of modifying a commercial manometer by either using a heavier than water fluid or by inverting them and using a lighter than water (LTW) fluid was also investigated but abandoned in favor of building a dual-fluid manometer.

In a dual-fluid manometer one of the fluids must be the process fluid. Page (1953) used a kerosine-water manometer for his work with pitorifices which gave a scale expansion of five. Preston (1972) mentioned modifying a micromanometer to a dual-fluid manometer but reported sluggish operation and zero-drift. Lin (1976) suggested the use of LTW silicone fluids in inverted dual-fluid manometers using glass tube which had been heat treated with silicone fluid.

Two double, inverted, dual-fluid manometers with leveling reservoirs were built to allow a total of four differential pressures to be measured simultaneously. Figure 5.3 shows a partial schematic of the manometers. A fill/vent funnel provides for the addition of LTW fluid and release of trapped air, a leveling jar allowed zeroing of one leg of a manometer, and traps prevented loss of the LTW fluid in the event of a pressure upset. Tubing was supported by an eight-foot piece of $3 \times 1.41 \times 0.17$-inch aluminum channel. The channel was scored at 50 mm intervals using a milling machine to make accurately spaced marks. A moveable frame supported the leveling jar on the channel; fine adjustments to the jar were made with a leveling nut to allow the lower meniscus to be zeroed at a marked interval. A steel rule was used to determine the height of the upper meniscus above the nearest scribed mark.

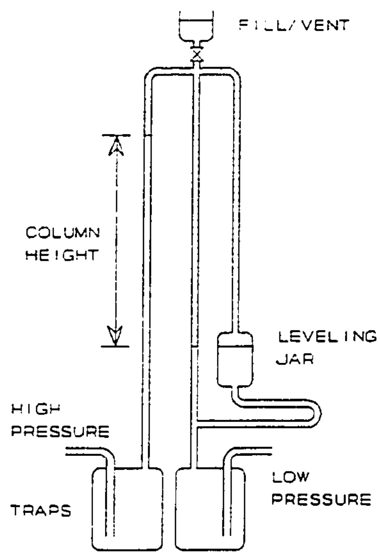


Figure 5.3: Dual-fluid Manometer and Traps

The dual-fluid manometer could be calibrated with the micromanometer as shown in Figure 5.2 by making measurements at two different pressures. An alternative to this is to use the difference in specific gravity (ΔSG) between the LTW fluid and water. If the density of a fluid is not known it can be found using a pycnometer and ΔSG calculated or ΔSG can be directly determined by measuring relative column heights in a U-tube:

$$SG_{LTWF} H_{1,2} \approx SG_W H_{2,3} \tag{5.2}$$

$$SG_{LTWF} \approx \frac{H_{2,3}}{H_{1,2}} \tag{5.3}$$

$$\Delta SG \approx 1 - \frac{H_{2,3}}{H_{1,2}} = \frac{H_{1,2} - H_{2,3}}{H_{1,2}} = \frac{H_{1,3}}{H_{1,2}} \tag{5.4}$$

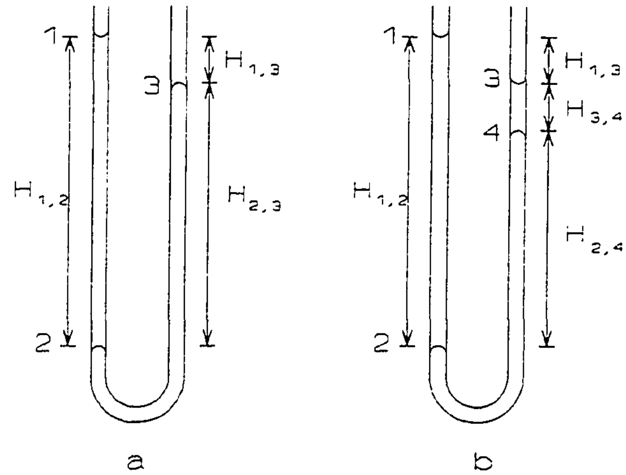


Figure 5.4: Determining Specific Gravity Difference by Balancing Fluid Columns

where SG is specific gravity, H is absolute value of the height difference between indicated positions, subscripts LTWF and W refer to LTW fluid and water, numeric subscripts refer to positions shown in Figure 5.4 and $SG_w=1$. If $H_{1,2}$ and $H_{1,3}$ are 2,000 mm and 200 mm ± 2 mm then ΔSG can be determined to within ± 1 percent (assuming no error due to surface tension effects):

$$\Delta SG = \frac{200 \pm 2}{2000 \pm 2} = 0.100 \pm 0.001 \tag{5.5}$$

so that in a dual-fluid manometer using this fluid, a differential of 50 ± 2 mm between the fluid column menisci would be 5.0 ± 0.25 mm WC. This is an error of five percent compared to measuring a WC differential directly which would give 5 ± 2 mm for an error of forty percent.

Because the fluids are immiscible it can be expected that there will be differences in the surface tension and the menisci at the air surface. A difference in the menisci shapes can indicate a relatively large error in $H_{1,3}$. In Figure 5.4a the LTW fluid is shown to be wetting the walls and the water is shown to be not wetting the walls. The effect will be a net additive force increasing $H_{1,3}$. This problem does not occur in a dual-fluid manometer because the difference in column heights is measured between identical interface conditions so menisci shapes are identical. This problem can be avoided in calibration by ensuring no water-air interface as shown in Figure 5.4b so that either the column heights are measured between identically shaped menisci or only differences between columns with the same bounding menisci shapes are used in the calculation. Following the reasoning above:

$$SG_{LTWF} H_{1,2} = SG_W H_{2,4} + SG_{LTWF} H_{3,4} \tag{5.6}$$

$$SG_{LTWF} = \frac{H_{2,4}}{H_{1,2} - H_{3,4}} \tag{5.7}$$

$$\Delta SG = 1 - \frac{H_{2,4}}{H_{1,2} - H_{3,4}} = \frac{H_{1,2} - H_{3,4} - H_{2,4}}{H_{1,2} - H_{3,4}} = \frac{H_{1,3}}{H_{1,2} - H_{3,4}} \tag{5.8}$$

If $H_{1,2}$, $H_{3,4}$, and $H_{1,3}$ are 2,000, 10 and 199 mm ± 2 mm then ΔSG can be determined to within ± 1 percent:

$$\Delta SG = \frac{199 \pm 2}{1990 \pm 4} = 0.100 \pm 0.001 \tag{5.9}$$

as in the above example. In both examples it is important to keep the value in the denominator as large as possible.

The LTW fluid is chosen for a desired ΔSG but viscosity is also a factor when considering response time. Assuming negligible acceleration time and laminar flow, the response time for a change in pressure can be estimated using the Darcy-Weisbach equation for head loss, Equation 3.16, and Poiseuille's Law, Equation 3.17:

$$H = \frac{32 L \nu V}{d^2 g} \tag{5.10}$$

where H is the height of an imaginary fluid with a density equal to the difference of the densities of the two fluids in the manometer, f is the Darcy friction factor, L is the length of tubing, V is the average velocity of the fluid, d is the ID of the tubing, and ν is the kinematic viscosity. Referring to Figure 5.5, assuming an instantaneous change in pressure (caused by adding a weight in the schematic example) that would be balanced if the meniscus moves from x=0 to x=h if the piston movement is negligible, and noting that the velocity in the pressure transmission line is greater than that in the manometer by D^2/d^2 Equation 5.10 can be written:

$$H = h - x = \left(\frac{L_T \nu_W D^2}{d^4} + \frac{L_W \nu_W + L_O \nu_O}{D^2} \right) \frac{32}{g} \frac{dx}{dt} \tag{5.11}$$

where the subscripts T, W, and O refer to the transmission line, water, and oil respectively. Separating variables and integrating from x=0 to x=0.95h:

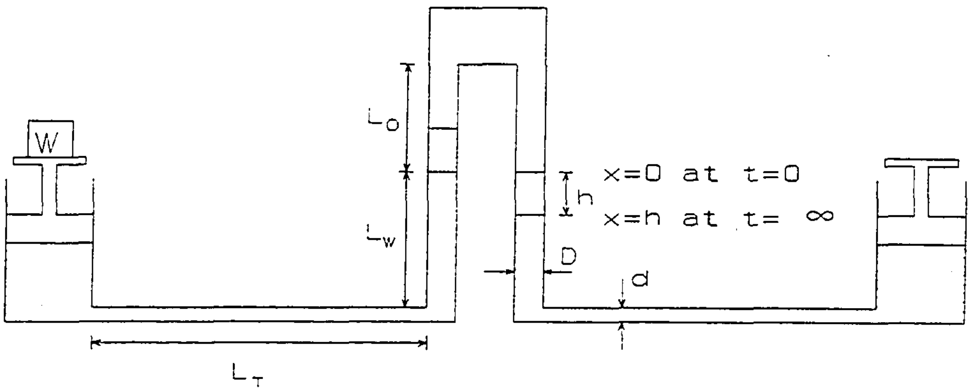


Figure 5.5: Estimating Response Time in a Dual-fluid Manometer

$$\int_{t=0}^{t_{0.95}} dt = \left[\frac{L_T\nu_W D^2}{d^4} + \frac{L_W\nu_W + L_O\nu_O}{D^2}\right]\frac{32}{g}\int_0^{0.95h}\frac{dx}{h-x} \tag{5.12}$$

$$t_{0.95} = \left[\frac{L_T\nu_W D^2}{d^4} + \frac{L_W\nu_W + L_O\nu_O}{D^2}\right]\frac{32}{g}\ln\left[\frac{1}{0.05}\right]$$

where $t_{0.95}$ is the time required for x to reach 0.95h. Using L_T=5 m, D=d=0.01 m, L_W=L_O=0.5 m, $\nu_W=\nu_O=10^{-6}$ m²/s,20ν_W, and g=9.83 m/s², the response time is found to be $t_{0.95}\approx0.6$ s. Setting $\nu_O=20\nu_W$ results in $t_{0.95}\approx1.5$ s, showing the impact of higher viscosity.

There is a potential advantage to using smaller diameter pressure transmission lines in that pressure fluctuations may be damped out, but this is at the expense of response time. Using d=0.005 m in the previous example the response time goes from 1.5 to 9 s. Pressure transmission lines were $^7/_{16}$-inch ID vinyl tubing with $^1/_{16}$-inch wall thickness. Spring clamps were used to close off inactive lines. Care was taken to prevent kinking by frequent inspection and by replacement of damaged lines. Bleeding of lines was done prior to every test run and lines were frequently inspected for air bubbles during runs. Transmission lines were also kept the same length so that both legs would respond at the same rate to system pressure changes.

Some properties and prices for LTW fluids are given in Table 5.1. Fuel oil is cheap, has low viscosity and offers a factor of five improvement in scale over an air-water manometer. Vegetable oil is cheap, non-toxic, and expands the scale by a factor of ten but it is very viscous. The Dow Corning 200® silicone oils are fairly low in viscosity, non-toxic, and expand the scale by up to twenty but they are relatively expensive. Initial experiments were made with fuel oil, vegetable oil, and Dow Corning 200® Fluid, 20 cs.

The shape and position of the meniscus depends on the properties of the fluids and the tube wall

Table 5.1: Specific Gravity and Viscosity of Potential Dual-fluid Manometer Fluids

Lighter-than-water Fluid	SG	Viscosity (cs)	$/pint
No. 1 Fuel Oil	≈ 0.8	≈ 2	
Vegetable Oil	≈ 0.9	≈ 70	
Dow Corning 200® Fluids (low viscosity polydimethylsiloxane polymers)	0.851	1.5	75.53
	0.872	2.0	75.53
	0.913	5.0	60.88
	0.935	10	60.88
	0.949	20	37.79

(Adam, 1940; Finn, 1986) and these can change with time, temperature, or contaminants. Effects can be minimized by increasing the radius of the manometer, but this can slow response time and may also make reading the levels more difficult.

Since $^7/_{16}$-inch diameter vinyl tubing was used for pressure transmission lines it was tried as manometer tubing also. One of the advantages of using that particular size was that it fit snugly over ½-inch OD copper tube; short pieces of copper tube were used for splicing and in the fabrication of tees and other fittings. Other advantages were that it was large enough to allow air bubbles to escape fairly easily in lower viscosity fluids and still small enough to not require large amounts of fluid or to make reading the menisci difficult.

Unfortunately, the surface properties of the vinyl tubing changed over time. Water would initially not wet the vinyl if it had been previously in contact with any of the oils, but after several days the surface would change and the water would wet it. This effect was investigated with a loop of vinyl tubing with water on the bottom and one of the oils on top. Air bubbles were eliminated by using a line off a tee which could be kept full as shown in Figure 5.6. Figure 5.6 also shows how the loop of vinyl tubing could be moved over supports to expose tubing to water after it had been in contact with oil and vice versa.

Untreated glass tube with a 12 mm ID was tried next but the menisci were not uniform along the length of the tube. Lin (1976) had suggested wetting glass tube with a 2 percent solution of silicone oil in methyl ethyl ketone then baking at 180°C for 30 minutes. This was tried several times but the treatment would not persist. Lin (1993) had never observed this because he had only done some preliminary work on the subject.

A chemist at Dow Corning suggested a caustic wash (saturated KOH in methanol) followed by a rinse with 5 percent solution of Dow Corning Z-6070 Silane in methanol and heating to 120°C for 30

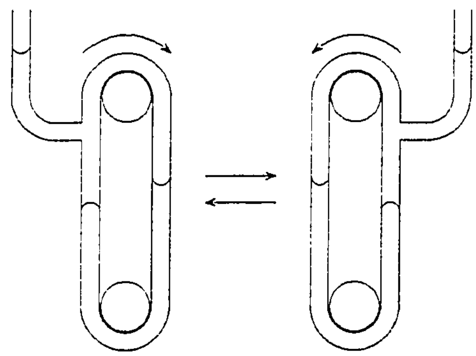


Figure 5.6: Affect of Meniscus Shape and Level by Last Fluid in Contact with Tube Wall

minutes for a stable hydrophobic coating (Pate, 1993). This was tried several times but, although the coating was much more stable, the properties of the coating were not uniform and this resulted in distorted shapes in the menisci.

Because of the difficulties discussed above and the need for an output suitable for a data logger, pressure transducers were purchased and the use of the dual-fluid manometers was discontinued. Simple manometers were used. These were made using vinyl tubing that was periodically replaced when problems with the menisci would develop. As such they were very helpful as a quick visual check on the differential heads being monitored. They were also used for calibration of the differential pressure transducers and in small circulation pump testing.

Levels were measured using either a 30-cm or a 150-mm steel rule.

5.1.2 Differential Pressure Gauges

Two Dwyer Instrument Co. (Michigan City, Indiana) Series 4000 Capsuhelic® differential pressure gauges with ranges of 0 to 0.5-inch WC and 0 to 3-inch WC were purchased for use in field test equipment. These gauges are suitable for line pressures to 500 psig, have an accuracy of ± 3 percent, and are relatively inexpensive and rugged but the response time was found to be up to 5 minutes in the lower range. According to the manufacturer, this slow response time is characteristic of the bellows type design at the lower pressure differentials. Diaphragm meters with faster response times are available but these are more expensive and were not purchased.

5.1.3 Pressure Transducers

A Bailey Controls Co. (Wickliffe, Ohio) model PTDD A111200 and two Validyne Engineering Sales Corp. (Northridge, California) model P305D differential pressure transmitters were purchased for testing of pitorifices. These sensors are based on a diaphragm positioning a magnetic core in a coil. Both models will withstand line pressures of 2,000 psi or more. A Micro Switch (Division of Honeywell;

Freeport, Illinois) model 16PC05DF differential sensor and a model 236PC150GW gage sensor were purchased for use in testing small circulation pumps and for recording water system pressures. These sensors are temperature compensated and relatively inexpensive. They are based on the flexing of a sensing diaphragm in a square silicone chip. The flexing causes a resistive imbalance in four integral piezoresistors arranged in a Wheatstone bridge configuration.

Both Rosemount Inc. (Eden Prairie, Minnesota) and Bailey offer differential pressure gauges which use diaphragms to move sensing units and which have response times under 1 second. The Rosemount model 3051C has an adjustable span of 0 to 25-inch maximum to 0.83-inch WC minimum. The Bailey model has an adjustable span of 0 to 2.4-inch maximum to 0.4-inch WC minimum. Both models have 4 to 20 mA output. The Bailey model was chosen because it costs less and was immediately available. It was used in the 0 to 2.4-inch [60-mm] WC range.

A stand was built for the Bailey transducer and fittings were made to allow connection of its sensing ports to vinyl tubing. The transducer assembly was installed at the base of a water manometer so that venting was continuous as shown in Figure 5.7a. This was possible because there were top and bottom ports for both high and low pressure taps and there were no constrictions in flow. A bypass line, which was normally kept pinched closed, was included to allow quick zeroing.

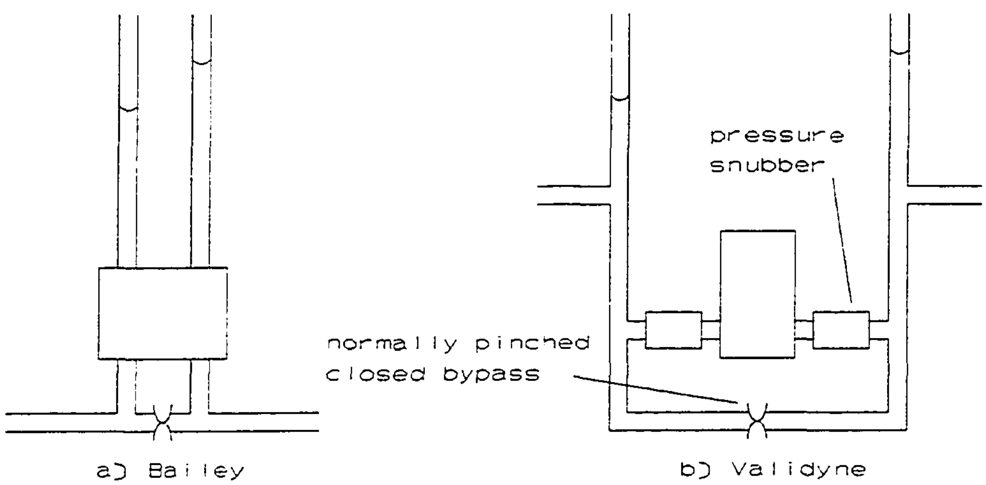


Figure 5.7: Mounting Arrangements for Differential Pressure Transducers

The Validyne transducers have an adjustable span of 0 to 3.5 inches maximum to 2.3 inches WC minimum with a 0 to 5 V output. They were used in the 0 to 3.5-inch [90 mm] WC range. They were purchased after it became apparent that the dual-fluid manometers would not be practical. They had not been located prior to the purchase of the Bailey transducer and were purchased because they were a little less expensive and it was thought that less signal conditioning would be needed. Unfortunately, signal conditioning proved to be troublesome due to the extreme sensitivity and fast response of the transducers.

Pressure snubbers were eventually added as part of the resolution of the conditioning problems.

The Validyne transducers were connected in parallel with water manometers. Pressure ports are on the side and venting of trapped air is done by loosening a vent screw. To prevent air from reentering once it had been bled off, the transducers were located below the level of the pressure transmission lines as shown in Figure 5.7b.

The Bailey and Validyne transducers were initially calibrated against the Dwyer micromanometer to ensure linearity. Test results are shown in Figures 5.8 and 5.9; testing started and ended at 0 inch WC to ensure no hysteresis effect. During use the transducers were calibrated at no differential and near full range before and after test runs against manometer columns to ensure logged values were accurate. Transducers were also occasionally connected to the same source to allow comparison of relative response and sensitivity to pressure fluctuations.

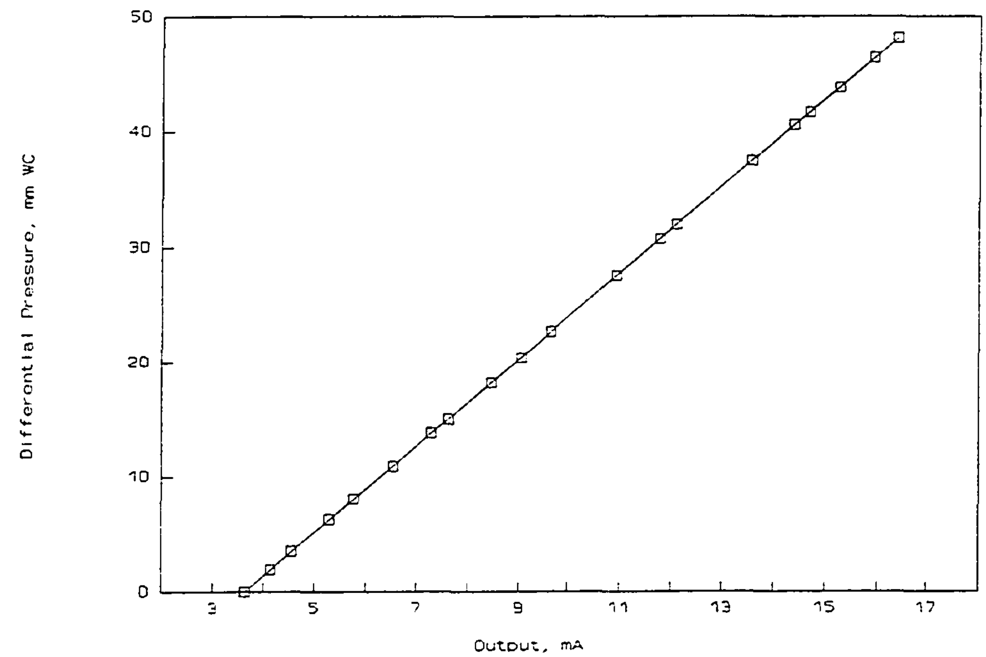


Figure 5.8: Calibration of Bailey Differential Pressure Transducer

The Micro Switch model 16PC05DF differential sensor has a 0 to 5 psi range and is suitable for use with liquids. Ports are 0.20-inch diameter and unvented. It is temperature compensated and has an unamplified linear 0 to 50 mV output signal with a 10 V input. When it was calibrated from 0 to 2.3 m WC and then back to 0 m WC, a 50 mm hysteresis was noted, about 10 times what was expected based on product literature. Because of this error, the sensor was not used.

A Micro Switch model 236PC150GW gauge sensor with a 0 to 150 psi range was used to monitor

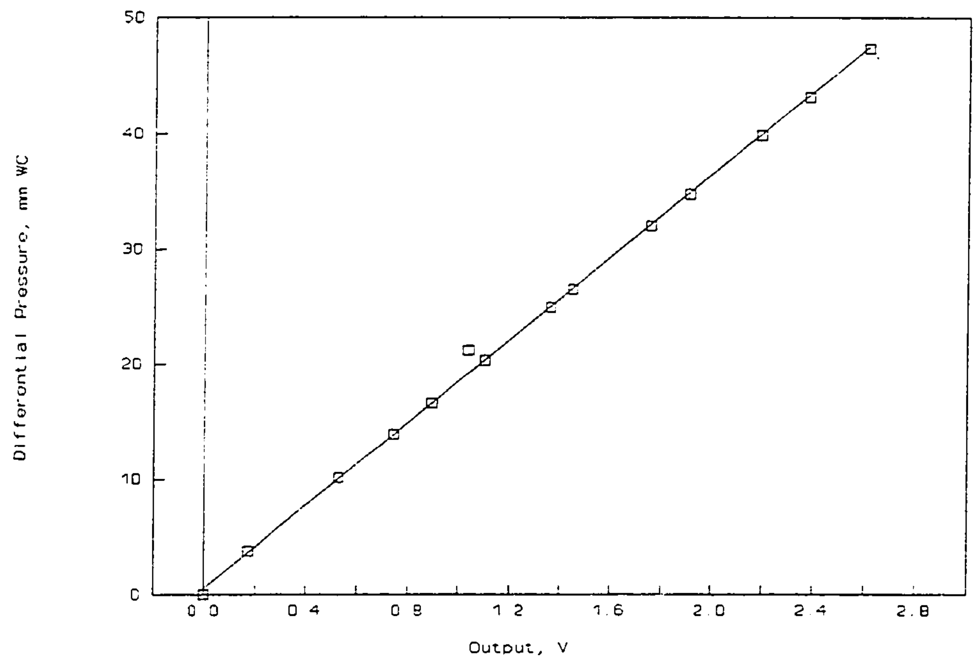


Figure 5.9: Calibration of Validyne Differential Pressure Transducer

water system pressures. The sensor has an unamplified linear 0 to 60 mV output signal with a 10 V input and was found to agree within ± 2 psi with a 0 to 150 psig bourdon gauge at midscale.

5.1.4 Piezometers

Two types of piezometers were made for use on nominal ¾ or 1-inch diameter Type K hard drawn copper water tube and one type for use on 4 and 6-inch diameter PVC pipe. The first two type were built onto a short length of tube with four ⅛-inch diameter holes equally spaced around the center. Tubing lengths varied depending on the application, but the rings were placed close enough to one end to allow inspection and deburring of the inside edges of the drilled holes. The tube was cut with a band saw rather than a tubing cutter to minimize deformation and the ends were either machined or sanded flush and then filed to remove burrs. The pieces were chucked in a ⅞ or 1⅛-inch collet held in a square receiver and a milling machine was used to drill the holes precisely. A deburrer was then used to clean the inside edges of the holes to ensure a flush finish.

The first type of piezometer was made by pressure forming a ring of copper which was then soldered over the pressure holes in the section of tube. Two rings were obtained by cutting 0.57-inch long pieces from the ends of a single standard sweat fit coupling. The rings were heated to about 1,100°F (a dull red heat in a dark room) to anneal them and then installed in a specially built die, the interior filled with oil, and a drop hammer used to hit a piston which caused the oil to expand the ring against the die.

Most times a single drop sufficed. The end of a ¾ to 2-inch long piece of ¼-inch OD soft annealed copper tube was then silver soldered to the ring. Initially the end of this tube was filed to fit the shape of the formed bulge but in later models the end was annealed, flared, annealed, then formed to a saddle shape by pushing it onto the bulge in the ring. While this entailed more steps it was a quicker and surer method of obtaining the close fit needed for silver soldering. A $^5/_{32}$-inch diameter hole was then drilled through the ring and the assembly aligned and soldered over the previously prepared piece of tube using lead/tin solder. Ideally, the ¼-inch diameter tube should be aligned between the ⅛-inch diameter piezo hole taps but it was found to be easier to position a piece for soldering if it was lined up with a piezo hole and held there with a small rod during soldering. Figure 5.10 shows the finished piezometer.

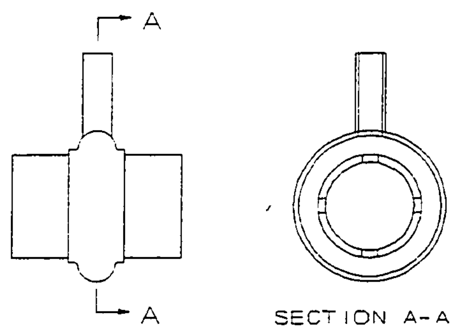


Figure 5.10: : First Type of Piezometer for Service Lines

The second type of piezometer was made by soldering ¾-inch long pieces of ¼-inch OD tube directly over each of the four piezo holes in the ¾ and 1-inch diameter tube. The ¼-inch diameter tube was flared and the flare deformed to match the OD of the tube. The prepared pieces were then held in place with a jig against the previously prepared ¾-inch diameter tube and soldered using lead/tin solder. Finally $^3/_{16}$-inch ID by $^1/_{16}$-inch wall vinyl tubing and three nylon tees were used to connect the holes as shown in Figure 5.11. This is very similar to a piezometer designed to minimize error due to pressure differentials between piezo holes (Blake, 1976).

The wall thickness for both ¾ and 1-inch diameter Type K water tube is 0.065 inch so the resulting ratio of piezo hole depth to piezo hole diameter is 0.52 compared to 1.5 which is normally recommended. The first type of piezometer required more fabrication effort and is less ideal in terms of geometry but is more rugged and can be used at higher pressures. The first type does not have symmetric piezo holes and passages so, with oscillating pressures, there could be more head loss going in one direction than the other. A literature search was made to ascertain the importance of hole geometry (Allen and Hooper, 1932; Emmett and Wallace, 1964; Franklin and Wallace, 1970; Freeman, 1941; Miller, 1983; Shaw, 1960). The error was judged to be insignificant given the oscillation rates observed and the

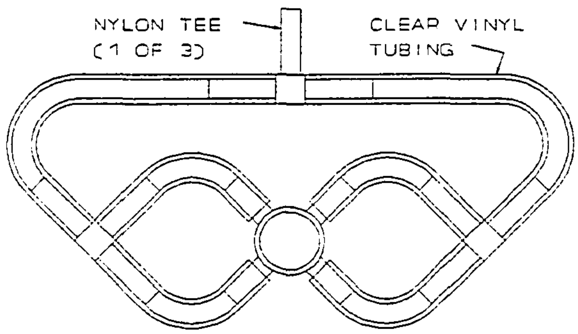


Figure 5.11: : Second Type of Piezometer for Service Lines

experimental results reported, so the first type of piezometer was used exclusively in testing.

Pressure connections were made by gluing $^{3}/_{16}$-inch ID by $^{1}/_{16}$-inch wall vinyl tubing inside $^{7}/_{16}$-inch ID by $^{1}/_{16}$-inch wall vinyl tubing using a succession of intermediate sizes. Piezometers were installed using standard compression or solder fittings. Solder fittings were sometimes sealed with electrical splice tape to allow easy disassembly.

Piezometers used on the 4 and 6-inch diameter PVC pipe were made by adding an outlet and frame to a rubber coupling gasket. The outlet was made from a 1¼-inch piece of ½-inch OD copper tube inserted into and soldered to a 1-inch square of 0.065-inch thick sheet copper bent to the inside radius of the gasket. A ⅜-inch diameter hole was drilled in the coupling gasket and the outlet inserted through it and sealed with silicone rubber sealant. A self-locking external retaining ring was slid down the tube to lock it to the gasket. A retaining frame was made from galvanized sheet metal to keep the gasket from deforming and leaking under pressure. The assembly was positioned over ⅛-inch diameter holes drilled around the circumference of the pipe with the outlet horizontal and positioned between the ⅛-inch diameter holes in the PVC pipe. Figure 5.12 shows a piezometer for 4-inch diameter PVC; the sectional view shows the ⅛-inch diameter hole rotated into view for illustrative purposes.

5.2 FLOW MEASUREMENT IN SERVICE LINES

A means of measuring and communicating flow rates ranging from 0.003 to 0.060 L/s [0.01 to 0.2 m^3/h; 0.05 to 2.00 gpm] in nominal ¾ and 1-inch diameter service lines was needed for full scale and field testing of pitorifice performance. Because head loss would lower the observed flow in the line, less than 1 mm [0.08-inch] WC pressure drop in the meter at low flow and less than 25 mm [1.0-inch] WC drop at high flow was desired.

A means of measuring flows to 0.2 L/s [3 gpm] was also desired for testing small circulation pumps. The following subsections discuss various methods of flow measurement that were investigated.

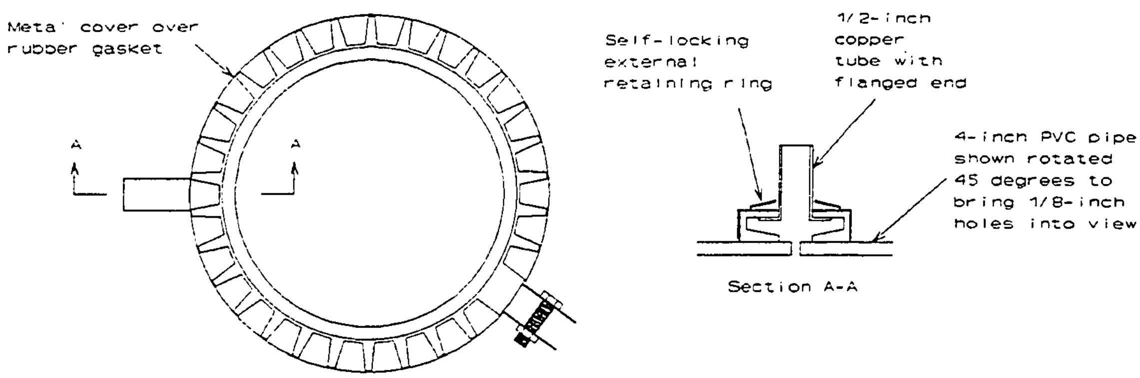


Figure 5.12: Piezometer for 4-inch Diameter PVC Main

5.2.1 Timed Fill

The small meter test facility at the Fairbanks Municipal Utilities System (MUS) water treatment plant has a calibrated tank which was used for some calibration and testing of equipment and techniques used for the service line flow measurements. However, because flows rates were usually very low, one or two 2-liter graduated cylinders were used in most situations. The cylinders had marks every 0.02 liters. A digital stop watch was used for timing. Accuracy was about ± 2 percent of the calculated flow for most cases, because at least 2 liters were sampled and fill times were usually 100 seconds or more.

5.2.2 Dye or Salt Velocity or Dilution Methods

Because of the low velocities in the service lines it is possible to time dye or salt movement in a very short length. The only successful field measurement of service line flow prior to this study was done this way in one test in Unalakleet; dye was injected into a 2-foot section of clear tubing to find a velocity of 0.2 fps (Murphy and Hartman, 1969). This method was also used in full scale testing done on pitorifices by the U.S. Army Corps of Engineers (Johnson, 1978) and the U.S. Public Health Service (Harris, 1987; Mauser, 1986; Spehalski, 1988). Instruments to detect concentrations are needed to extend the range and remove human subjectivity from the process. The mean transit time is then obtained by using the center of gravity of the curve of concentration versus time. An ISO Standard is available for tracer methods (ISO, 1974). See also Hooper (1940 and 1962) and John (1976).

In the dilution method a dye or salt is continuously injected at a known mass flow rate into the pipe and the concentration in the water is determined after a suitable mixing distance. The flow rate of the water is then determined by a mass balance. Fluorescent dyes in concentrations invisible to the eye are commonly used for this purpose. A fluorometer is then used to measure concentrations (Turner Designs, 1990). See also Kilpatrick (1968). Dye and salt velocity methods were dropped from consideration because of the time required to make a measurement.

5.2.3 Rotameter (Variable Area Meter)

Rotameters are well suited for low flow rates, they are inexpensive, a 10 to 1 turndown is standard, and they can be installed directly after a fitting. However, the head losses for all the commercially available meters which had the appropriate flow ranges were greater than the maximum allowed. Modification of a commercially available rotameter was considered; modification may be guided by the theory (Bean, 1971; Holman, 1971) and confirmed by calibration. The shape, density, and relative area of the float are important. A thin disk float is nearly independent of viscosity changes in the fluid and a dense float is less affected by density changes in the fluid. If the float area is larger than the annular area, the float position is nearly linear with flow rate.

A Fischer & Porter Co. (Warminster, Pennsylvania) precision bore Flowrater® model rotameter was provided by the MUS. It was modified by replacing the brass inlet and outlet pieces which had ½-inch female pipe thread (FPT) outlets at right angles to the tube with specially machined fittings and ¾-inch diameter copper pipe stub-outs. The pressure drop from the float can be approximated by the weight over the largest cross-sectional area. The float diameter was a maximum of 0.508 inch and it weighted about 0.3 ounces so the pressure drop was about 2.5 inches WC, much too great for service line flow measurements. The range was 0.05 to 0.45 gpm but this may have been extended by replacement of the stainless steel float. A replacement float could not have a larger maximum diameter and if its diameter was significantly less than the original, some means of centering it would have been needed. A guide wire through the float, guide prongs on the float, or modification of the float so that it spins were considered. Due to accuracy limitations and the modifications required, this method of flow measurement was not pursued further. The rotameter was used during initial head loss testing of service line fittings to allow quick adjustment of flow.

5.2.4 Ultrasonic Transit-time Meters

Ultrasonic meters were considered because they can be strapped on the outside of a pipe and do not restrict the flow. A Controlotron Corp. (Hauppauge, New York) model 994P4G type 5 clamp-on transit-time ultrasonic flow meter was tested on nominal 1-inch diameter steel pipe. Problems were observed with zero-drift and accuracy was poor. After discussions with factory engineers it was decided that a clamp-on style would not be appropriate for such low flows and a flow-through model would be too expensive compared to other options.

5.2.5 Calibrated Head Loss Devices

Because the pressure drop in a meter would have to be taken into consideration in many cases anyway, just measuring the head loss from a flow restriction was considered. Two different devices were built, the first used a valve for the flow restriction and had to be calibrated for every setting every time it was used and the second had an orifice and could be used with a previously prepared calibration curve.

Figure 5.13 shows a schematic of the first device, termed a flow test pipe, in operation on a service line. The first view shows the service line, the second shows the union between the two service lines opened and the flow test pipe connected to it with two 2.5-foot lengths of ¾-inch diameter reinforced vinyl hose, and the last view shows calibration. The flow test pipe consisted of a 2.5-foot run of ¾-inch diameter Type K copper water tube and sweat fittings, a union and an elbow, a 0.5-foot run of copper tube with a ball valve used for shutting off and throttling flow, a second elbow and union, and a second 2.5-foot run of copper tube and sweat fittings. Thermistors, indicated by a circled "T", were installed at sweat fit tees located 55 inches apart from each other measured along the tube run with the ball valve. Piezometric rings were located 31.5 inches apart and were connected through filters to two Dwyer Capsuhelic differential pressure gauges with 0 to 0.5-inch and 0 to 3-inch WC ranges. Pressure transmission lines were arranged and valved so that either side could be the high pressure side and the flow test pipe was symmetric so that flow could be in either direction.

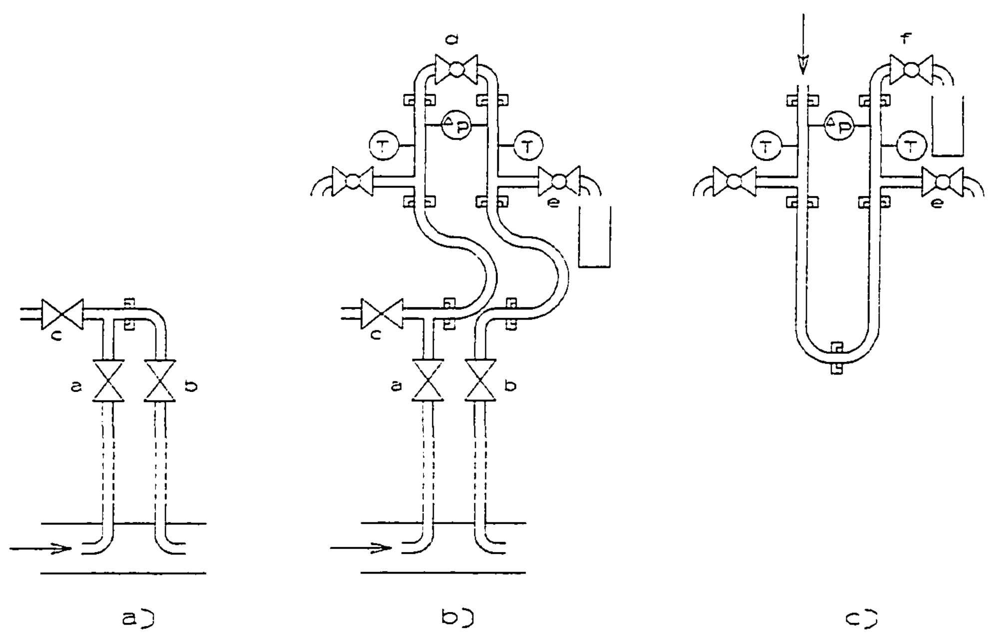


Figure 5.13: Using a Flow Test Pipe to Measure Service Line Flow

To collect flow and pressure drop data on a service line, the service line union was isolated and opened, the elbow rotated, and hoses attached to connect the flow test piping. The ball valve was set at positions from full open to full closed. At each position the differential head across the valve is noted, then the gate valve on the return leg of the service loop was shut and water was withdrawn from the test piping through a tee (valve "e" in Figure 5.13) at a rate that duplicates the differential head loss across the ball valve. This rate of flow was determined by timing the filling of a graduated cylinder.

The flow that would result if the flow test piping were joined at the two piezometric taps was determined by plotting points showing the pressure drop as a function of the flow, fitting a curve, and extrapolating to the x-axis. The flow in a service line without the flow test pipe was determined by adding the system curve for the hoses and piping between the hoses and the piezometric taps to the fitted curve and extrapolating to the x-axis. To find this hose and flow test pipe system curve the hoses were joined as shown in Figure 5.13c and head loss as a function of flow was measured and plotted as shown in Figure 5.14.

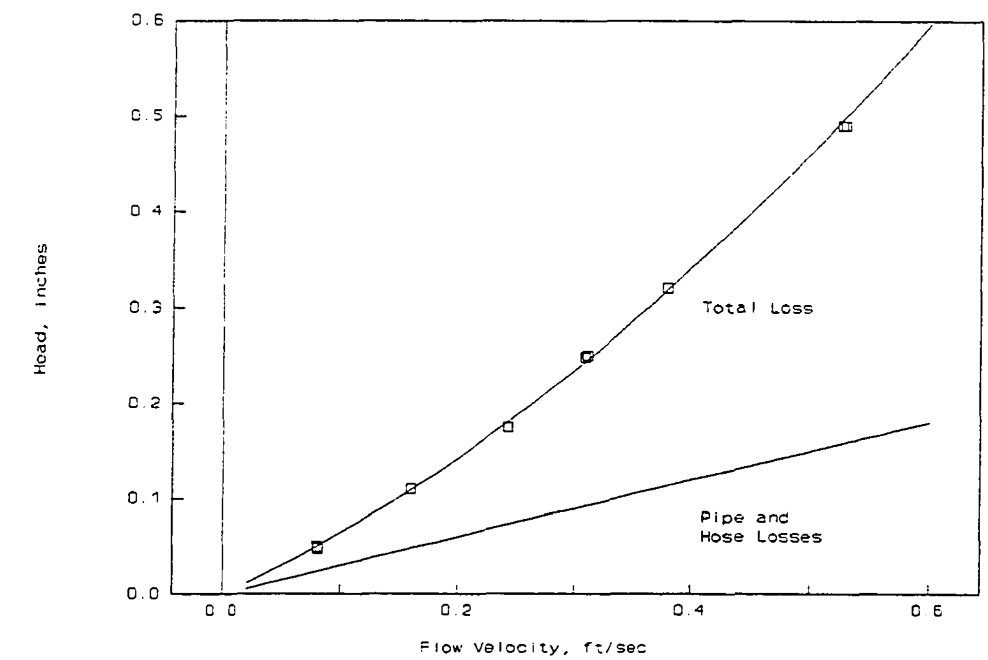


Figure 5.14: Flow Test Pipe and Hose Head Losses

The testing was done at 45 psig, which served to hold the hoses round. It was assumed that the diameter change with pressure between 45 psig and pressures in various water systems was negligible in the reinforced hose. A curve was fitted to these points by first assuming a laminar flow system curve for the total run length of 8 feet, then finding the difference between the data points and the laminar piping system curve. This difference was assumed to be due to fitting losses. These losses were primarily due to constrictions in the hose-to-male pipe thread (MPT) fitting adapters. The fitting loss coefficient was then calculated ($K=2gH/V^2$) and plotted against Reynolds number as shown in Figure 5.15. These points were then fitted with a curve and the function for K from this curve was used to fit the points in Figure 5.14. These points might have been directly fitted with a curve, but assuming a dependence on Reynolds number allows the result to be used at different temperatures with more confidence.

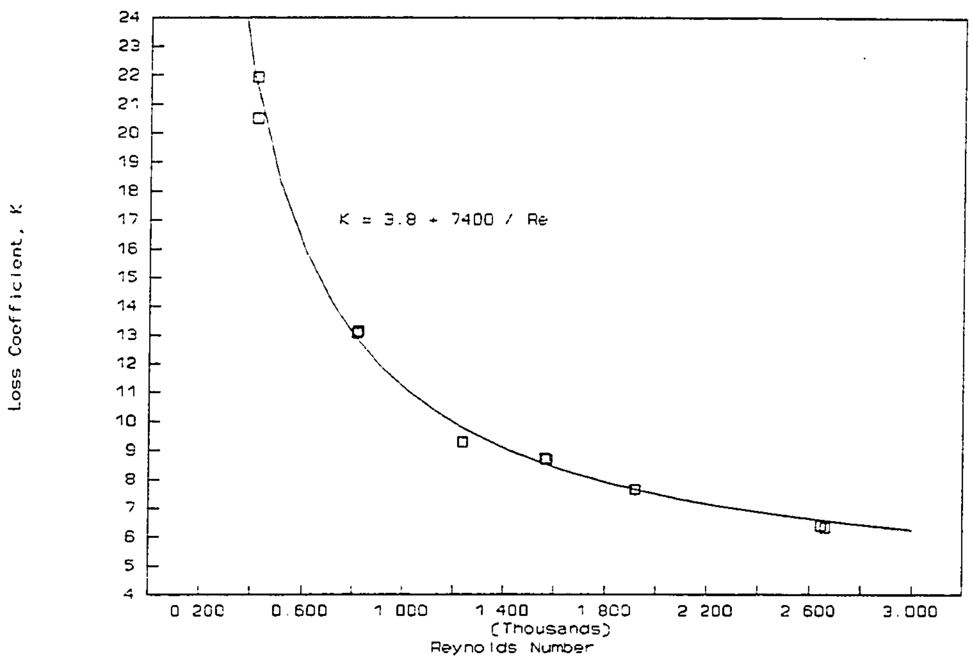


Figure 5.15: Flow Test Pipe Fitting Loss Coefficient Versus Reynolds Number

The sum of the hose and flow test pipe system curve and the service line test curve yields the service line performance curve which, when extrapolated to the x-axis, gives the predicted flow for the service line during normal operation. An example is shown in Figure 5.16. The different values shown for shut-off head illustrates one source of error, changing pressure differentials during the test. Fluctuating system pressures also caused a bouncing in both levels which made differences difficult to determine at times. Another source of error is the reestablishment of flow through the throttled ball valve after the gate valve on the return leg of the service line is closed and the water is started to be bled off through the tee. The flow loss through a partly closed valve can exhibit a hysteresis effect, so throttling was always done from the same direction when making a pair of readings.

Because the performance curve can be approximated by a straight line it is possible to estimate service line flow using only the shutoff head and one other point. An inexpensive and simple device using this principle was developed for quick checks on flows and for use by water system operators. The first version consisted of a ¾-inch ball valve installed in a length of tube which contained a ½-inch long brass plug with a ¼-inch diameter hole. The second and final version of what was by then called a bypass manometer (BPM) consisted of a 1-inch ball valve. A ½-inch long brass plug with a ¼-inch diameter hole was soldered into the ball. Stab fittings were installed in the valve body on either side of the ball as close together as practical. One-foot lengths of ⅜-inch ID vinyl tubing with $^{3}/_{32}$-inch wall thickness were attached to serve as manometer legs. These were terminated in a specially machined fitting which allowed the

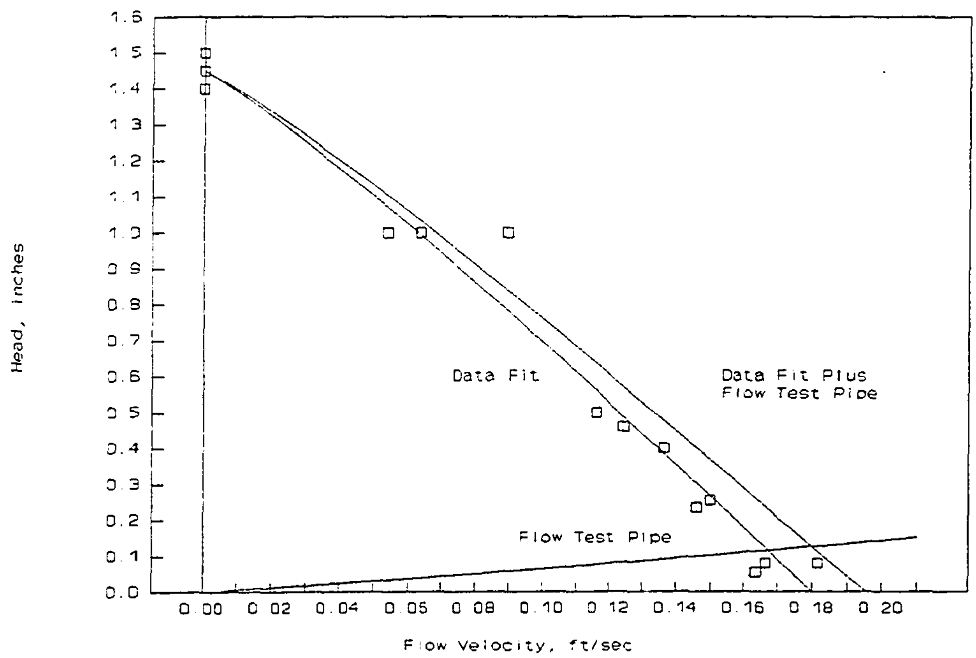


Figure 5.16: Actual Service Line Flow Determination with Flow Test Pipe

tubing to be installed close together and contained snifter and bleed valves to allow air to be added or vented as needed. Installing the manometer legs close together allowed them to be easily held together for taking readings and kept the air volume above the water levels to a minimum which in turn reduced the bounce and made the differential head easier to read. One-inch diameter reinforced vinyl tubing with hose coupling adapters threaded for pipe thread allowed quick installation to pipe fittings.

Figure 5.17 shows a calibration curve fit to points obtained with two different BPMs in which flows were in both directions. The temperature varied from 50 to 59°F [10 to 15°C] but no temperature dependence effect was apparent for that range. United States customary system (USCS) units were used because the devices were ultimately intended for field use by water system operators in Alaska and because the differential pressure gauges used in the calibration read in inches of WC.

Figure 5.18 illustrates the procedure for using the BPM. The service line piping is first opened up and the individual lines purged at a high flow rate to ensure they are unplugged and to clear sediment from them. Then the BPM is installed in the service line piping and both legs of the service line are individually purged of air. Sufficient air is then added through the BPM snifter valve to allow the manometer to work. Shutoff and bypass head readings are then taken with the ball valve closed then with it open. These are plotted on the y-axis and on the BPM curve as shown in Figure 5.19. A line drawn through theses points to the x-axis yields the predicted service line flow.

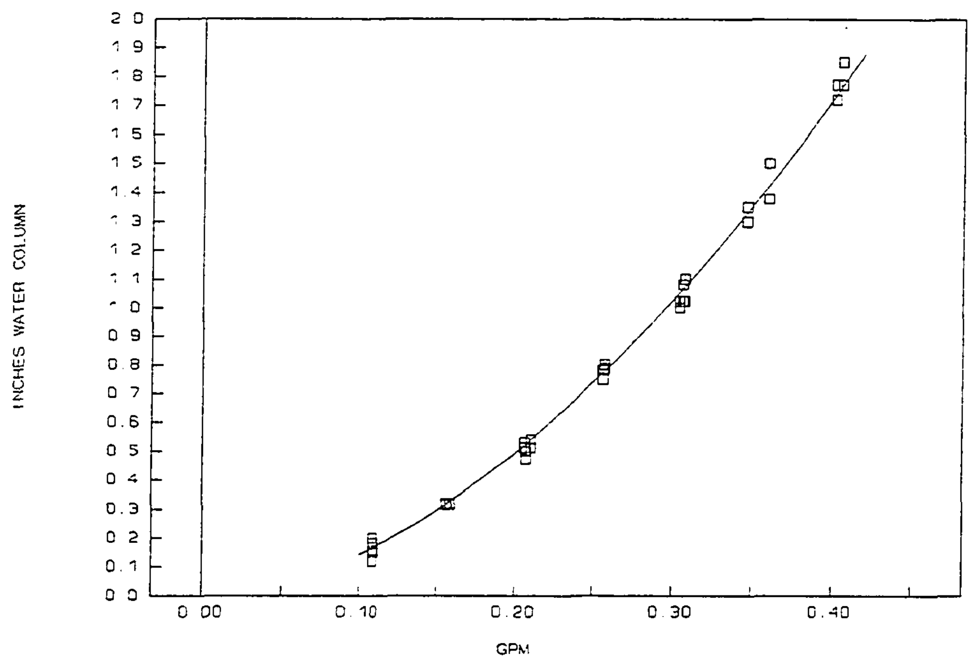


Figure 5.17: Calibration of the Bypass Manometer

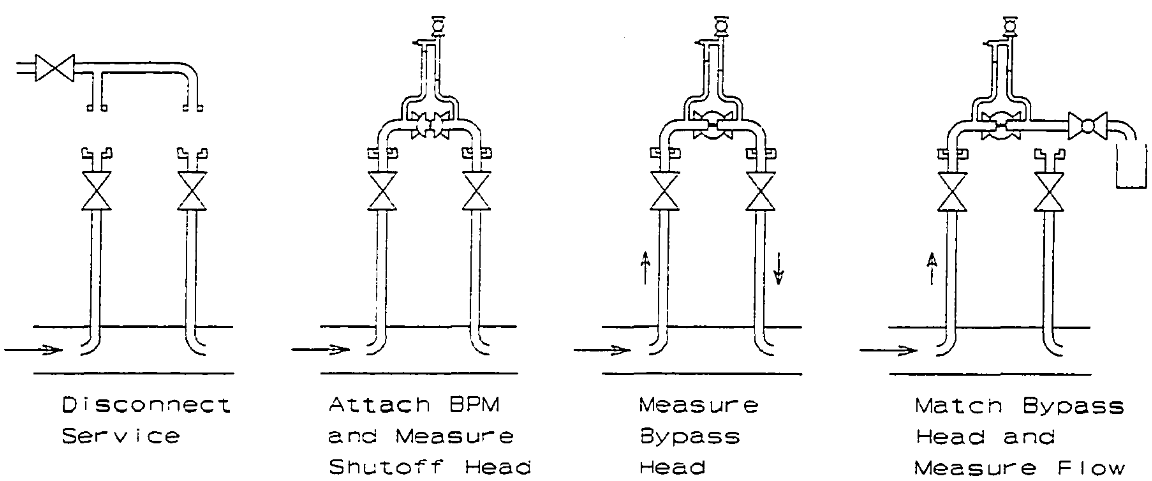


Figure 5.18: Bypass Manometer (BPM) Used to Determine Service Line Flow Rate

The curve fit for the BPM system curve was $H=8.92*Q^{1.8}$. By using a slightly less accurate fit of $H=11.1*Q^2$, a formula for service line flow can be written as:

$$Q \approx \frac{0.3H_{SO}\sqrt{H_{BP}}}{(H_{SO}-H_{BP})} \tag{5.14}$$

where Q is in gpm, H is in inches of water, and the subscripts SO and BP refer to the shut-off and bypass conditions.

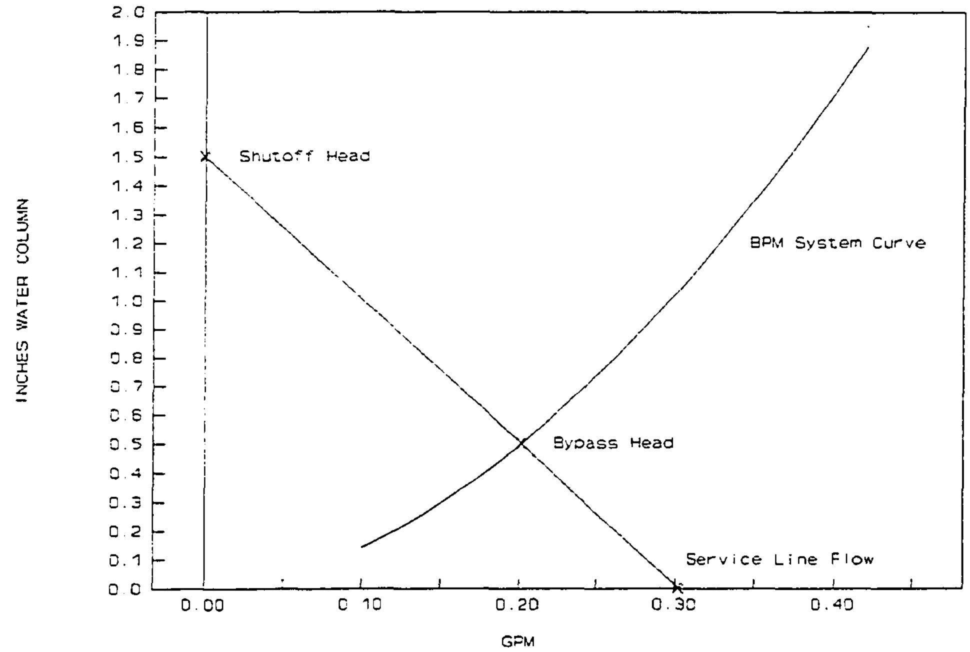


Figure 5.19: Bypass Manometer (BPM) System Curve with Service Line Plot

Instead of using the calibration curve, the flow rate can also be determined by withdrawing water off one side or the other through a throttling valve and determining the flow, Q_{BP}, which corresponds to the previously measured H_{BP} as shown in the final view in Figure 5.18. The unrestricted flow can now be estimated using the linear relationship $Q \approx H_{SO}Q_{BP}/(H_{SO}-H_{BP})$.

The bypass manometer, unlike the flow test pipe, requires the assumption that the performance curve of the pitorifice/service line system can be approximated with a straight line. This is a reasonable assumption if there is laminar flow and the service line head losses are large compared to fitting and pitorifice losses. The bypass manometer is not practical when pressure fluctuations are large or when pressure differentials are small. The flow test pipe was used in the first part of the study and the BPM was used throughout the study.

5.2.6 Thermistor Flowmeter

Thermistors can be used to measure flow velocities indirectly by monitoring their heat loss rate to the fluid. This is done by either sensing the temperature of a heated probe relative to that of the surrounding fluid or by using the self heating property of a thermistor in different ways. Intek, Inc. (Westerville, Ohio) manufactures thermistor flowmeters but they were too expensive for the project. A literature search verified that a thermistor flowmeter meeting the low flow rate requirement could be built relatively inexpensively (Briggs-Smith and Piscitelli, 1981; Forstner and Rützler, 1969; Graf, 1991;

Katz and Shoughnessy, 1987; LaBarbera and Vogel, 1976; Rasmussen, 1962; Vepřek, 1963). Prototype models were constructed but abandoned when funding became available for a magnetic flowmeter.

5.2.7 Magnetic Flowmeter

Magnetic flowmeters operate by sensing the electric field which develops when a conductor, water, moves through a magnetic field (AWWA, 1989; Bean, 1971; Christensen, 1989). An ISS Clorius International A/S (Ballerup, Denmark) model 5VPD Magnetic Flow Meter was evaluated and initially used for the study and a model 3VPD was later purchased for the study and used during most of the study. MUS provided both Clorius meters. Both meters have the same physical dimensions and differ only in their flow ranges. The model 3VPD has a 0.00125 to 3.0 m^3/h [0.00035 to 0.833 L/s; 0.0055 to 13.2 gpm] range and the model 5VPD has a 0.0025 to 5.0 m^3/h [0.00069 to 1.389 L/s; 0.011 to 22.0 gpm] range. Both meters have an accuracy of ± 1 percent of range. The 3VPD was purchased with passive 4 to 20 mA output.

The head loss for laminar flows was determined for the 5VPD using the Dwyer differential pressure gauges and the flow test pipe with two 6-foot lengths of hose to connect the meter. The head loss attributable to the meter alone was then determined by subtracting the curve fit for the losses determined when the meter was replaced with a 2-inch length of 1-inch diameter brass pipe from the curve fit for the losses with the meter included. The results are shown in Figure 5.20 and are applicable to the model 3VPD also because it has identical geometry. The losses in the meter alone are equivalent to a 3.0 to 3.5-foot length of 0.745-inch ID tube.

In nominal ¾-inch diameter Type K copper water tube, a velocity of 1 fps is equivalent to 1.36 gpm. Flows induced by pitorifices in this size service line are usually in the laminar region which extends up to about 0.4 fps or 0.6 gpm. A correction can be made to a reading taken with a Clorius meter in the same way as was done with the flow test pipe.

The head loss for turbulent flows was determined for the 3VPD meter using a manometer and piezometers on either side of the meter. Figure 5.21 shows the head loss in meters of WC plotted as a function of the dimensionless quantity $f \cdot V^2/d \cdot 2 \cdot g$ where f is the Darcy friction factor, V is the velocity that would occur in nominal ¾-inch diameter Type K copper water tube, and d is the tubing ID. The slope of the line through the points is 9.1 feet [2.77 m] and this can be regarded as the equivalent length of the meter in terms of 0.745-inch ID tube in turbulent flow as shown by rearranging the Darcy-Weisbach equation:

$$H = L\frac{fV^2}{d2g} \qquad (5.15)$$

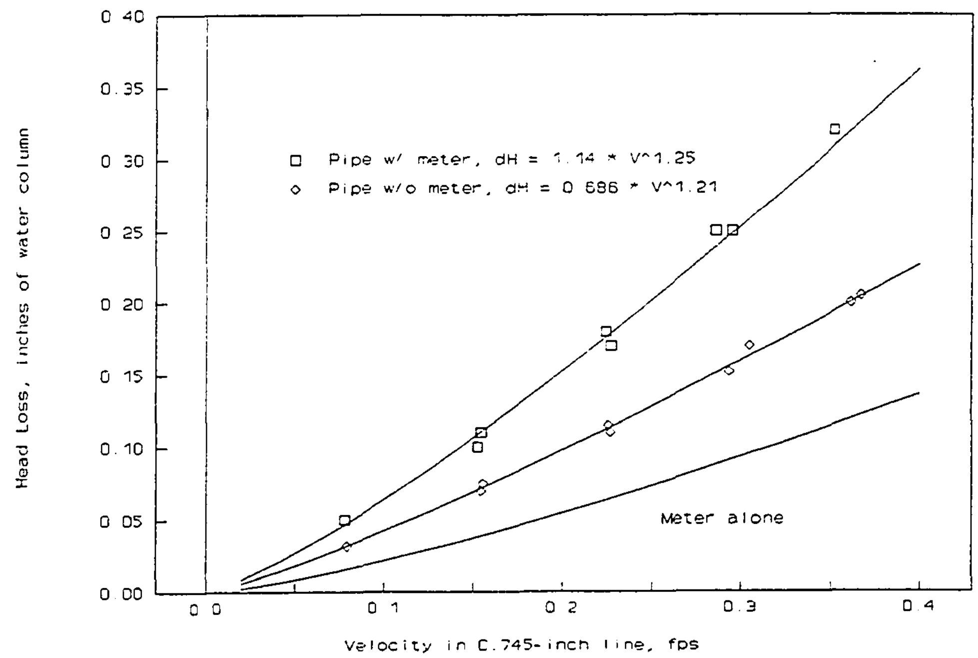


Figure 5.20: Head Loss for Clorius 5VPD for Laminar Flow

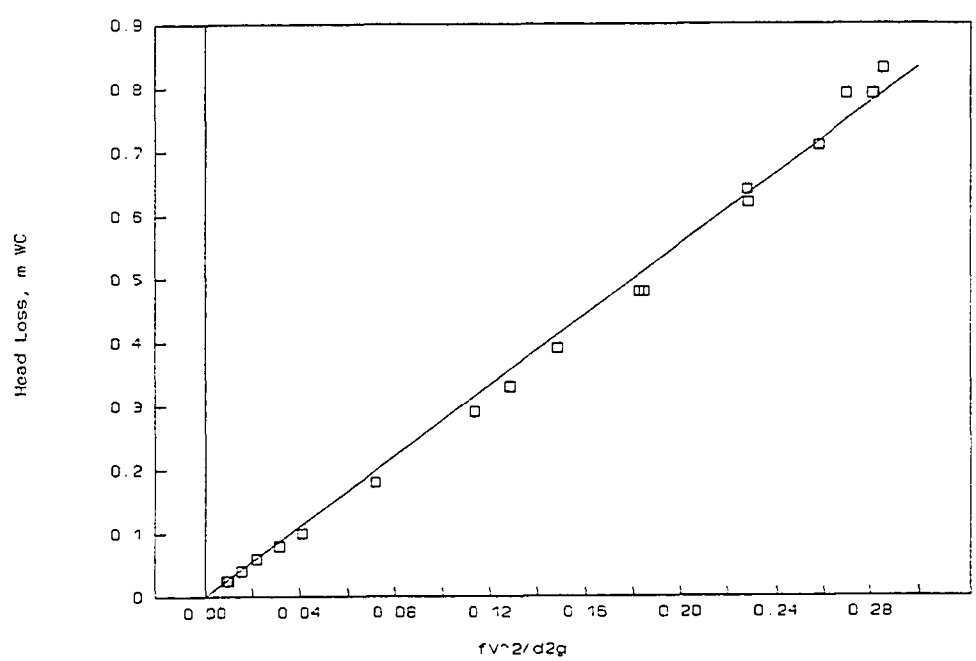


Figure 5.21: Head Loss for Clorius 3VPD for Turbulent Flow

Calibration over the range of interest showed the meter accurate to the limits of calibration, which was about ±1 percent when timing the filling of graduated cylinders. Subsequent calibration was done with the data logger prior to using the meter in test runs.

5.3 FLOW MEASUREMENT IN MAINS

A means of measuring and communicating flow velocities ranging from 0.1 to 0.9 m/s [0.3 to 3.0 fps] in the full scaled test sections of the main was needed. These velocities result in flow rates of 1 to 17 L/s [3 to 62 m^3/h; 13 to 274 gpm] in the nominal 4 and 6-inch diameter schedule 40 PVC pipe used in the tests. A Clorius model 75 Magnetic Flow Meter was purchased with funding provided by MUS. The Clorius model 75 has a 0.0375 to 75.0 m^3/h [0.0104 to 20.8 L/s; 0.165 to 330 gpm] range with an accuracy of ±1 percent of full scale. It was factory calibrated and supplied with passive 4 to 20 mA output.

A weigh or volume tank was initially considered because of the high accuracy of these methods of flow measurement and because such a facility could be subsequently used by MUS for meter testing. These options were rejected because of cost and complexity. A weigh tank would require load cells and an amplifier costing a minimum of $2,000. A volume tank would not give continuous flow readings and would require construction of a flow diverter. Both methods would require frequent restarts. Finally, some remodeling at the water plant would have been required to locate a tank in the test area.

Orifice plates were also considered because they are relatively cheap and, when carefully machined and installed, can be accurate to ±1 percent (ASME, 1989). Unfortunately, the rangeability is only 4:1 so a minimum of 3 different orifice plates would have been required to cover the flow range. Although the plates and flanges are relatively inexpensive, one or two differential pressure gauges allowing 4 to 20 mA output would also be needed. Flow nozzle, venturi, averaging pitot tube, propeller, and ultrasonic meters were also considered and rejected for high cost, lack of range, or lack of accuracy.

Two techniques for measuring flows in mains in field testing, dye dilution and an ultrasonic meter, were also considered. Dye dilution flow measurement could potentially be used to measure flows in mains in field studies by continuously injecting dye at a known mass rate at one service and taking samples at a downstream service for determination of concentration in a fluorometer. Because free chlorine will affect the dyes, an adjustment might be required to account for residence time. This concept was not actively pursued because of cost and complexity.

Clamp-on ultrasonic transit time flow sensors were also considered but access to mains with adequately long straight runs was limited and the coal tar epoxy or cement linings in steel and ductile iron pipes can prevent or render readings inaccurate. Had funding levels permitted, however, an ultrasonic sensor would have been purchased because, with appropriate transducers, it could have been used on service lines also. Other methods of flow measurement, such as pitot tubes, turbine meters, or paddle wheel sensors, were not considered because they required drilling a hole in the main.

5.4 TEMPERATURE MEASUREMENT

Temperature had to be measured because the viscosity of water varies with temperature. Also, accurate measurement of small changes in temperature was required for heat loss observations. Briggs (1990) discusses use of thermocouples and thermistors with data logging equipment. Type T copper/constantan thermocouples were initially used with a Fluke 2100A Digital Thermometer and a Fluke 2150A Multipoint Switch Unit for monitoring water temperature for service line head loss testing. However, thermocouples are accurate to only ± 0.5 F° and the instruments did not have outputs suitable for automatic data logging.

Betatherm Corp. (Shrewsbury, Massachusetts) 5K3A1 thermistors were initially used in thermistor probes made by gluing a thermistor with bare wire leads between ¼-inch thick pieces of acrylic then cutting and turning the acrylic to a $^{3}/_{16}$-inch OD for insertion through a compression fitting. The tip of the thermistor was exposed to the water and the epoxy coating could not be relied upon to withstand emersion. Betatherm 10K3A thermistors were subsequently purchased in stainless steel housings with ⅛-inch male pipe thread (MPT). Temperature was calculated using the Steinhart-Hart equation with the constants provided by Betatherm:

$$T = \left[A + B\ln R + C(\ln R)^3\right]^{-2} - 273.15$$

$$\begin{array}{rl} \textit{Where} & T = {}^\circ C \\ & R = \text{ohms} \\ \textit{for Betatherm 10K3A:} & A = 1.129241\text{E}{-3} \\ & B = 2.341077\text{E}{-4} \\ & C = 8.775468\text{E}{-8} \\ \textit{for Betatherm 5K3A1:} & A = 1.28745\text{E}{-3} \\ & B = 2.357394\text{E}{-4} \\ & C = 9.5052\text{E}{-8} \end{array} \tag{5.16}$$

The thermistors were calibrated relative to each other when small differences in temperature were being monitored. Proper operation was verified by ice bath. Horowitz and Hill (1989) describes an amplifier that could have been used to measure small changes in temperature about a reference temperature using a thermistor and assuming linearity over a small range.

5.5 ESTIMATING SERVICE LINE LENGTH AND CONDITION

In most cases service line lengths in the field were determined by measuring the distance to the main and the condition of the lines were deduced from the flow rates observed. To supplement and verify these determinations, two other approaches were considered. In the first method the service lines are isolated from each other at the house and a differential pressure gauge installed between them. Water is then withdrawn at a known rate from one line and the line length is calculated using an assumed hydraulic roughness for the pipe, the observed pressure difference, and an assumed total fitting loss. If the line length is known, the condition of the pipe can be judged based on the discrepancy between the known and calculated line lengths.

In the second method, the line length is calculated from the period of oscillation of water in the line. The rate of damping of the oscillation gives a qualitative indication of the condition of the line. The pair of service lines can be regarded as equivalent to two one-arm U-tubes which are in turn equivalent to a single U-tube, as shown in Figure 5.22, provided they oscillate in the same direction and the trapped air is not allowed to expand and compress. In a pitorifice driven service with water flowing in the main, the effect of the pitorifices is the same as having the reservoir levels of the one-arm U-tubes at different elevations.

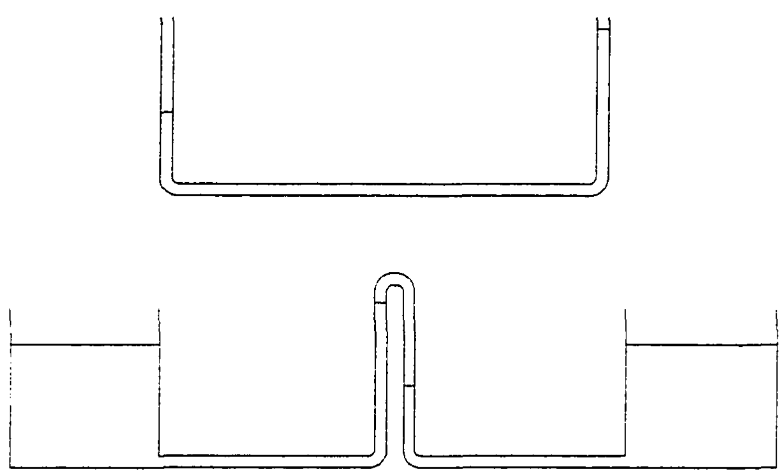


Figure 5.22: Single U-tube is Equivalent to Two One-arm U-tubes

If clear reinforced vinyl hoses with the same ID as the service lines are attached to the service lines, air can be trapped and the menisci observed. A ball valve and a snifter valve inserted between the hoses allow the water to be put into motion. After installation of the hoses the service line gate valves are opened and trapped air is pressurized. Excess air is bled out the snifter valve and the ball valve is closed. Then air is pumped into or bled out of one leg to achieve an unbalanced level and the ball valve is rapidly opened. Several oscillations can usually be observed before they are damped out too much for observation.

Because the period of oscillation is usually from 5 to 10 seconds it is relatively easy to time. After assuming negligible effects due to column ends, tube curvature, film adhesion, and purging, Letelier and Leutheusser (1976) gives the period of oscillation, T_0, as:

$$T_0 = \frac{2\pi}{\omega_n \sqrt{1-\zeta^2}} \tag{5.17}$$

where ω_n is the natural frequency of the flow system and ζ is the constant friction coefficient (also called the damping ratio). The natural frequency is given by:

$$\omega_n = \sqrt{\frac{2g}{L}} \tag{5.18}$$

where L is the length of the fluid column. The constant friction coefficient is given by (Jaysinghe et al., 1974):

$$\zeta = Real\left[\sqrt{\frac{i}{N_{Va}}}\,\frac{I_1\left(\sqrt{iN_{Va}}\right)}{I_2\left(\sqrt{iN_{Va}}\right)}\right] \tag{5.19}$$

where "Real[..]" means "the real part of," i is the square root of negative one, N_{Va} is the Reynolds number of oscillatory flow (also called the Valensi number), and I_n denotes the modified Bessel function of the first kind of order n. The Valensi number is given by:

$$N_{Va} = \frac{\omega_n r^2}{\nu} \tag{5.20}$$

where r is the tube radius and ν is the kinematic viscosity of the fluid. The modified Bessel function of the first kind and of order n is given by (Kreyszig, 1967):

$$I_n(x) = i^{-n} J_n(ix) = \sum_{m=0}^{\infty} \frac{x^{2m+n}}{2^{2m+n} m!\,\Gamma(m+n+1)} \tag{5.21}$$

where J_n is the ordinary Bessel function of the first kind of order n and Γ is the Gamma function.

Assuming nominal ¾-inch diameter Type K copper water tube with a radius of 0.00946 m, a range of lengths from 10 to 100 m, and a range of temperatures from 0 to 10°C and kinematic viscosities from 1.787×10^{-6} to 1.307×10^{-6} m^2/s, the range for the Valensi number is from 22 to 96. Equation 5.19 can be approximated in that range by:

$$\zeta = 1.67 N_{Vd}^{-0.6446} \tag{5.22}$$

Combining Equations 5.17, 5.18, 5.20, and 5.22 yields:

$$T_0 = \pi \sqrt{\frac{2L}{g}} \left[1 - 2.79 \left(\frac{L v^2}{2 g r^4} \right)^{0.6446} \right]^{-0.5} \tag{5.23}$$

Figure 5.23 is a graph of this equation for the conditions assumed above. Note the small effect of temperature. The plotted data point represents the test result for a one-arm U-tube test using a 20.8 m length of tube. The last term of Equation 5.23 can be dropped without more than five percent error.

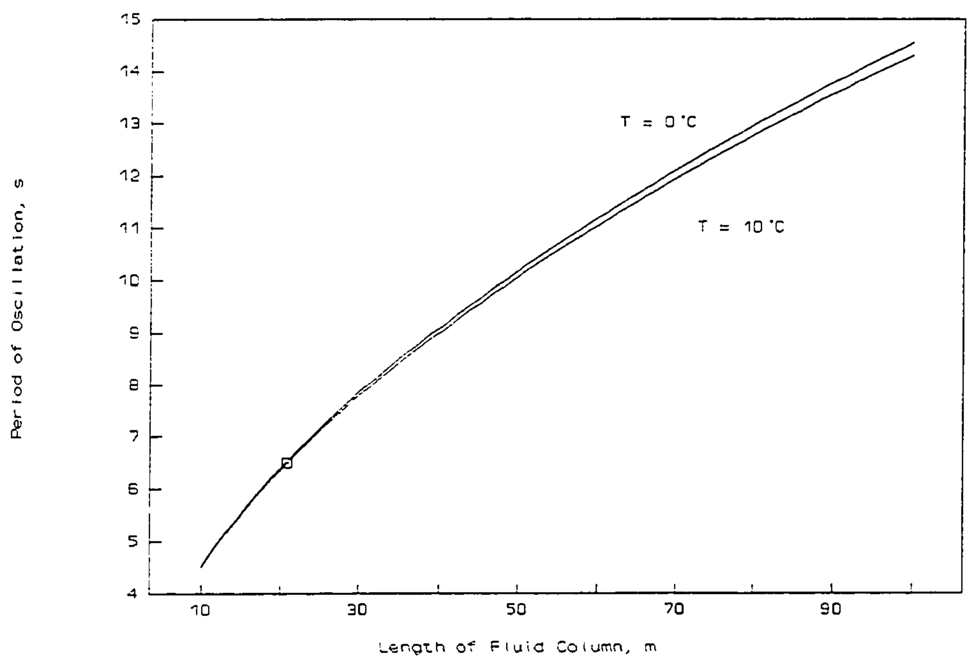


Figure 5.23: Period of Oscillation Versus Fluid Column Length

5.6 ELECTRONIC TEST EQUIPMENT

Resistance, voltage, amperage, and power all had to be measured for data logging and calibration purposes. A John Fluke Mfg. Co., Inc. (Everett, Washington) Model 87 True RMS Multimeter was used for most measurements. A Fluke Model 80i-kW Current/Power Probe was used with the multimeter for large pump motor power use determinations. A 100 amp single phase General Electric Type I-70-S Watt · hour meter was used for most other pump motor power measurements but it required timing of the disk movement for watt determination and a complete revolution at 10 watts took over 10 minutes.

A faster disk speed was obtained by using a three phase watt · hour meter with the inputs wired in series but this was still not satisfactory. A very simple direct readout power meter was finally constructed with watt ranges down to 10 watts by modifying a schematic provided by National Semiconductor (Graf, 1985; National Semiconductor, 1994a). A second simple power meter was subsequently identified which would have allowed data logging (Woodward, 1994). Using a more sophisticated design (National Semiconductor, 1994b) or purchasing a wattmeter were considered but the cost and complexity were too great. When the wave form was sinusoidal, the power factor could be determined using an oscilloscope and the power factor then used to convert VA to true power.

5.7 SIGNAL CONDITIONING AND DATA LOGGING

A model 7000 Starlog Macro data logger with a field termination strip was loaned by FPE Roen Engineers, Inc. (Fairbanks, Alaska). This data logger can be programmed to log on demand or at set intervals with operation monitored by a computer terminal. Data logging was usually done at 1 Hz because the data logger could not be monitored at the computer terminal at faster rates and also because the Clorius meters only refreshed every 0.4 seconds.

The data logger was first used to record voltage across the thermistors. A synchronized 5 volt source on the data logger supplied voltage across a thermistor in series with a nominal 10,000-ohm 5 percent tolerance wire wound resistor which had a resistance of 9,960 ohms. The 5 volt source was programmed to turn on 0.1 second before the data logger sampled to allow the thermistor to stabilize. The voltage across the thermistor was logged as a 0 to 5 volt analog voltage input with a resolution of 1.25 mV and an accuracy of ± 0.1 percent of range. The data logger was programmed to calculate the temperature by first solving for the thermistor resistance, R:

$$R = 9960 \frac{V}{5 - V} \tag{5.24}$$

where V is the voltage drop measured across the thermistor. This calculated resistance was then used in Equation 5.16.

The best resolution can be obtained by selecting the series resistor so that there is a maximum change in the voltage over the temperature range of interest. Noting that:

$$V_2 - V_1 \propto \frac{R_2}{R_S + R_2} - \frac{R_1}{R_S + R_1} \tag{5.25}$$

where V_1, V_2, R_1, and R_2 are the thermistor voltage drops and resistances at the temperature extremes of interest and R_S is the resistance of the series resistor. The maximum voltage difference can be determined by differentiating Equation 5.25 with respect to R_S, setting it equal to zero and solving for R_S:

$$R_S = \sqrt{\frac{R_1 R_2^2 - R_1^2 R_2}{R_2 - R_1}} \tag{5.26}$$

The Clorius flow meters had passive 4 to 20 mA outputs. These were supplied with a 14 V DC power supply in series with a nominal 25-ohm 5-percent tolerance wire wound resistor. The voltage across the resistor was measured as a 0 to 500 mV analog voltage input. At currents between about 8 and 18 mA, a 3,000 Hz signal with 500 mV amplitude was observed. This high frequency signal was not a problem with conventional readout and metering devices but it caused problems with the data logger which sampled over very small time intervals. A 100 μF electrolytic capacitor was installed across the 25-ohm resistor to serve as a bypass filter with a low impedance shunt for the high frequency signal across the resistor.

The Bailey differential pressure transducer had a passive 4 to 20 mA output and was supplied with a 14 V DC power supply in series with a nominal 25-ohm 5-percent tolerance wire wound resistor. The voltage across the resistor was measured as a 0 to 500 mV analog voltage input and a capacitor shunt was not required.

The Validyne differential pressure transducers were supplied with 14 V DC power and had 0 to 5 V DC output. These transducers were found to be extremely sensitive. Vibrations from the flow in the pipe were in the subaudible 1 to 10 Hz range as expected from vortex shedding. The signal output was limited to 0.5 mA at 5 V and a simple bypass filter would have been inappropriate. A low-pass filter could have been made by adding a 100,000-ohm resistor in series to the data logger and a 16 μF capacitor to ground, but changes in the logger impedance would affect calibration. Hydraulic damping seemed more appropriate because the noise was not truly electronic, but was actually an overly sensitive following of the signal itself.

Omega Engineering Co. (Stamford, Connecticut) Model PS-4E pressure snubbers with porous metallic elements with an average opening of 0.0013 [0.033 mm] were installed on the inlets to the Validyne transducers. This slowed the response time but eliminated the problems with excess sensitivity without compromising accuracy. Figure 5.24 shows the Validyne response to pressure change compared to that for the Bailey. When logging on one second intervals, there was still some variation at high differentials with vibration so the average of five readings over five seconds was logged. The first logged value was routinely discarded because the data logger would return an instantaneous reading and not an average when logging was initiated. Also, because of the slow response to changes, logging was only initiated after a 30 second wait.

When the upper range of a Validyne transducer was exceeded, the output would exceed 5 volts

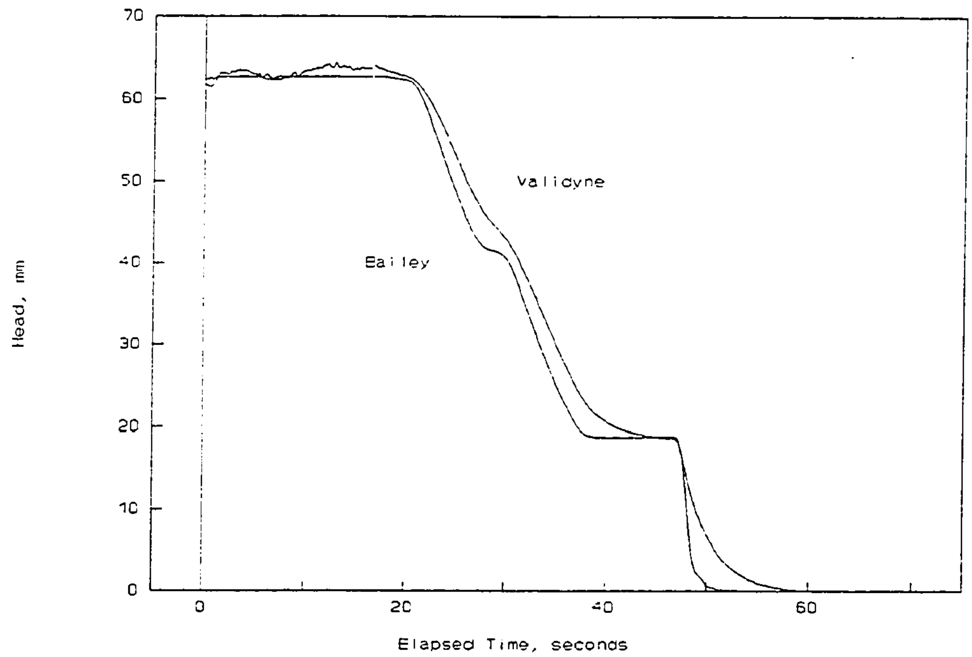


Figure 5.24: Outputs for Bailey and Damped Validyne Transducers

and this was found to cause the data logger to malfunction. The Validyne transducers were therefore disconnected whenever the pressure was out of range.

To aid in interpreting recorded data, an index voltage was also logged. This voltage was obtained by adjusting a 10-turn variable resistor to allow different voltage values to be sensed. When varying the insertion depth of pitorifices, the index voltage was made to corresponded to the insertion depth in centimeters relative to the center of the pipe. Dates and times were also logged. All logged data were imported into a spreadsheet at the conclusion of each test and comments added to aid in future interpretation. Data were usually promptly reduced and a draft report written at the same time.

Logging was controlled by opening and closing a switch. A stopwatch was used to keep track of the logging interval and the number of logged data points. Handwritten notes were also kept during test runs.

5.8 MISCELLANEOUS

Templates were made for measuring the outer bend radii of pitorifices. The templates spanned the range of 1.00 to 3.75 inches in 0.25-inch increments. Other instruments used included a dial caliper with a 6.5-inch range and 0.001-inch gradations, a dial gauge with a 5-mm range and 0.01-mm gradations, a strobe light, and a reflection type tachometer.

CHAPTER 6: PITORIFICE SHAPES

The first pitorifices were made by Mueller Co. (Decatur, Illinois) by bending a copper tube, then cutting the bend off flush along the straight edge of the tube and machining the tube in a lathe down to its original outside diameter. This is the bend, cut and turn (BC&T) shape shown in Figure 6.1. Ford Meter Box Co. (Wabash, Indiana) manufactures pitorifices by bending a copper tube only part way resulting in the incomplete bend, cut and turn (IBC&T) shape. Both companies have manufactured pitorifices using just the bend and cut steps and omitting the turning as in shapes B&C and IB&C. Note that turning decreases the projected area and enlarges the lower half of the opening. Pitorifices have also been made by cutting a straight tube on an angle then bending the tip over (C&B) or just by cutting the tube on an angle (C).

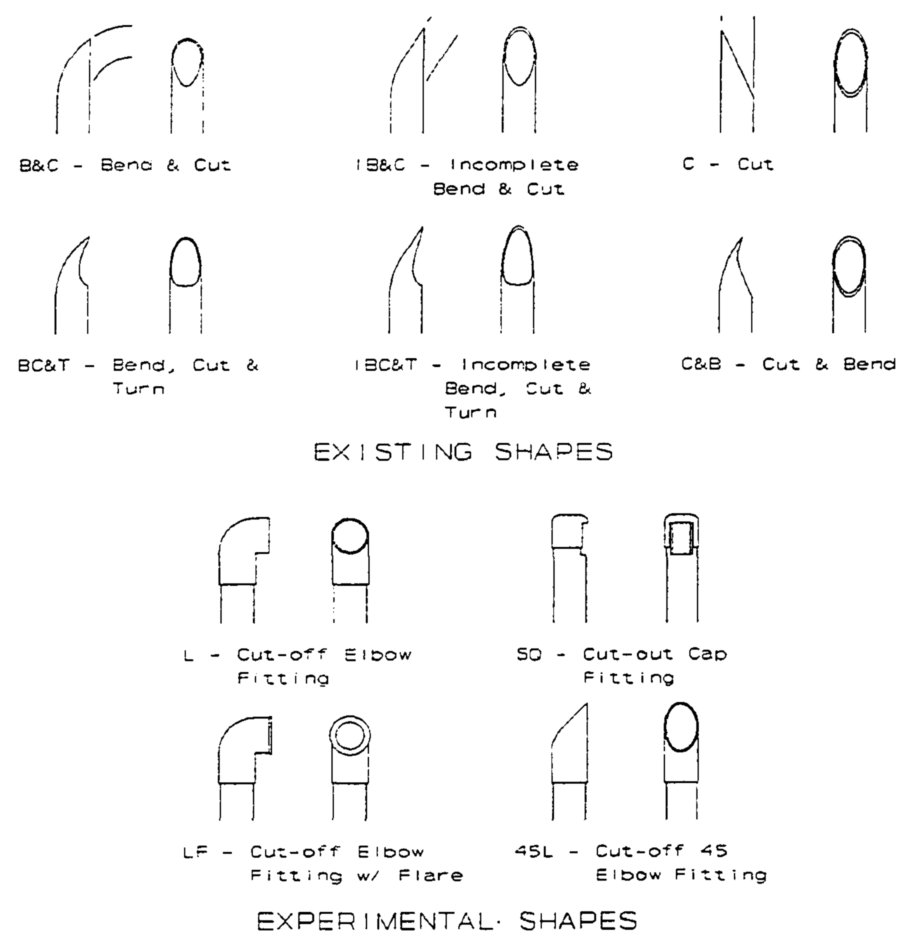


Figure 6.1: Pitorifice Shapes Investigated

Also shown in Figure 6.1 are the shapes of pitorifices specifically made for this study. These were made using standard copper sweat fittings to ensure uniformity, particularly for different sizes of the same shape. All the pitorifices shown in Figure 6.1 are scaled to a tube OD of 0.875, the OD for nominal ¾-inch diameter copper water tube, to allow easier comparison of shapes. All the shapes shown were evaluated in the study, with the exception of B&C; only one B&C specimen was located, Mueller no longer manufactures pitorifices, and a suitable copy could not be made with the resources available.

6.1 PITORIFICE CONSTRUCTION TECHNIQUES

Pitorifices have been made from a variety of sizes and thicknesses of tube bent to a variety of radii and degrees of completeness. Both Mueller and Ford bent tubing without causing it to deform significantly, Mueller typically changing the diameter less than 1 percent and Ford less than 4 percent. The tubing used has a minimum of 99.9 percent copper and has excellent cold bending properties. Ford built a special jig to do the bending and presumably Mueller did something similar. Both companies apparently did final machining in a lathe although rotating the bent and cut tube against a belt sander will also produce good results.

Pitorifices with the C&B shape used in the study were obtained from the contractor who was installing them in 1991 in Fairbanks when the last of the wood stave pipe installed in 1953 was being replaced. He obtained them from Western Utilities Supply Co. (Seattle, Washington) which in turn subcontracted them to a former employee who had been making pitorifices this way since before Mueller stopped making them. He cut the tube at 65° then bent the tip over.

The C shape pitorifice was the only one of the existing shapes which was made specifically for the study. It was made by cutting a tube at 45°.

Shapes BC&T, IB&C, IBC&T, and C&B were all salvaged from new or used pitorifices. The BC&T pitorifices date from 1953 and were salvaged during the 1991 construction job mentioned above, the others came from stockpiles of new and used fittings. They were all removed from the corporation stops and fitted to short lengths of nominal ¾ and 1-inch diameter Type K hard drawn copper water tube by machining the two pieces to fit into each other and soldering. In the case of the nominal ¾-inch pitorifices this resulted in a mismatch in the ODs.

Pitorifices were made specifically for use in the study. The smaller than normal models allowed determination of the effects of the relative sizes of the pitorifice and the main. Different shaped pitorifices were also fabricate to determine if they might be cheaper or easier to make or if more effective.

Several bending techniques were explored. Copper tubes were annealed by heating to a dull red, then bent by hand over a pipe or in a hydraulically operated jig. Tubing bent by hand would partially collapse; it could be reshaped in a vise but the results were not as uniform as those made by Mueller or

Ford. A tight fitting spring was slid over the outside of some tubes in an effort to limit deformation but this was not very successful for the very small bend radii required. It may have been possible to fill a tube with a low melting point alloy so that it would bend like a bar without collapsing but this method would result in high cost pitorifices. Small Parts Inc. (Miami Lakes, Florida) sells alloys of bismuth which can be used for this purpose. A 58/42 bismuth/tin alloy melts at 281°F [138°C] and costs about $30/pound. About 3 pounds would be needed to fill a 1-foot length of nominal 1-inch diameter copper water tube. Cheaper alloys and alloys having lower melting points are available, but these contain lead and would be inappropriate for use with potable water.

Bending a tube in a jig results in less deformation but the jig available was designed to bend 1.0625 inch OD tube and no other jigs were located in machine shops in Fairbanks. A piece of 1.000-inch OD tube was bent in the jig using a piece of rubber to shim it so it was snugly held in place. The rubber did not prevent the tube from partially collapsing when it was bent, but the tube was successfully reshaped in a vise then turned in a lathe. This specimen was not used in the performance testing but it was used in a prototype of an HDPE side fusion spool which is described in Appendix C. Attempts to bend smaller diameter tubes in the jig were unsuccessful. Modification of the jig was considered but not done because it would not lead to as inexpensive way to make pitorifices.

A variety of standard sweat wrought copper pressure fittings were considered. The more common types are available in a wide range of sizes. Those chosen, a 90° close rough ell cup-by-cup (C×C), a 45° fitting ell ftg.×C, and a tube cap, are all available from Nibco Inc. (Elkhart, Indiana) for nominal ⅛, ¼, ⅜, ½, ⅝, ¾ and 1-inch diameter tube with ODs of 0.250, 0.375, 0.500, 0.625, 0.750, 0.875, and 1.125 respectively. A 0.45-inch OD pitorifice in a 6-inch diameter main can be used to model a 0.75-inch OD pitorifice in a 10-inch ID main because the ratio of the pitorifice projected area to the main cross-sectional area is the same in both cases.

Shape L was made by soldering a 90° ell to a nominal ¾-inch diameter tube, then cutting 0.47 inch off the end of the fitting with an abrasive cut-off saw. Shape LF was made by soldering a short piece of copper tube which had been flared to an OD of 1.075 inches into a shape L resulting in a 0.05-inch lip.

Shape SQ was made by cutting a one-quarter section from a cap to within a distance of ¼D of the end where D was the tube OD (or the cap ID). The cap was then fitted to the end of a tube and the tube scribed so that it could be cut to result in an opening with a height of D. After cutting the tube the cap was soldered on. This resulted in a shape that was reproducible in the different sizes. A cheaper and easier construction technique may be to solder a cap on a tube and drill or cut a round hole in place of the rectangular hole.

Shape 45L was made by soldering a 45° fitting ell to the end of a tube and cutting it off flush along the soldered end of the fitting with an abrasive cut-off saw. A 45° fitting ell was used instead of a standard ell because a cup end does not extend up to the cut-off point. A 45° fitting ell was not locally available in a nominal ⅜-inch size, so a 45° ell C×C was used with a short piece of tube soldered in the cup to extend it past the cut-off point. Nominal ⅜-inch diameter hard drawn copper water tube was not locally available so brass tube was used for that size.

6.2 PITORIFICE GEOMETRY

A mathematical solution is useful in creating profiles and sectional views and in determining projected outside and opening areas. The following subsections give solutions for shapes B&C, BC&T, IB&C, and IBC&T assuming no distortion due to bending and assuming cutting and turning flush with the unbent portion and allowing the possibility of an offset to account for cutting or turning the end off at a radius greater than that of the original tube. Refinements can be made by assuming an elliptical cross-section in the bend to account for collapsing and bulging and assuming a tilt in the stem to account for bending of the stem. These are, however, relatively minor effects and they were not judged to be worth the extra computational effort.

6.2.1 Intersection of a Torus and a Plane

The shape of the tip of a B&C type pitorifice can be approximated by a torus intersecting a plane. Figure 6.2 shows an x-y plane view of a quarter section of a torus with major and minor radii of R and r_i intersecting an x-z plane at $y=R-r_o$. A solution of z in terms of x allows a face-on view to be constructed of the open area of a B&C type pitorifice with an inside radius of r_i. If the pitorifice is made by cutting flush along the unbent portion, as is normally the case, r_o represents the outside radius of the tube used for the pitorifice, otherwise r_o is the offset of the cut from the center line of the unbent portion of the tube. By setting $r_i=r_o$ the solution for the outside surface can be found from the general solution.

Referring to Figure 6.2 it can be seen that the limits of x are:

$$x_{min} = \sqrt{(R-r_i)^2-(R-r_o)^2}$$
$$x_{max} = \sqrt{(R+r_i)^2-(R-r_o)^2} \tag{6.1}$$

At any point P common to both the torus and the plane:

$$z = \pm\sqrt{r_i^2-d^2} \tag{6.2}$$

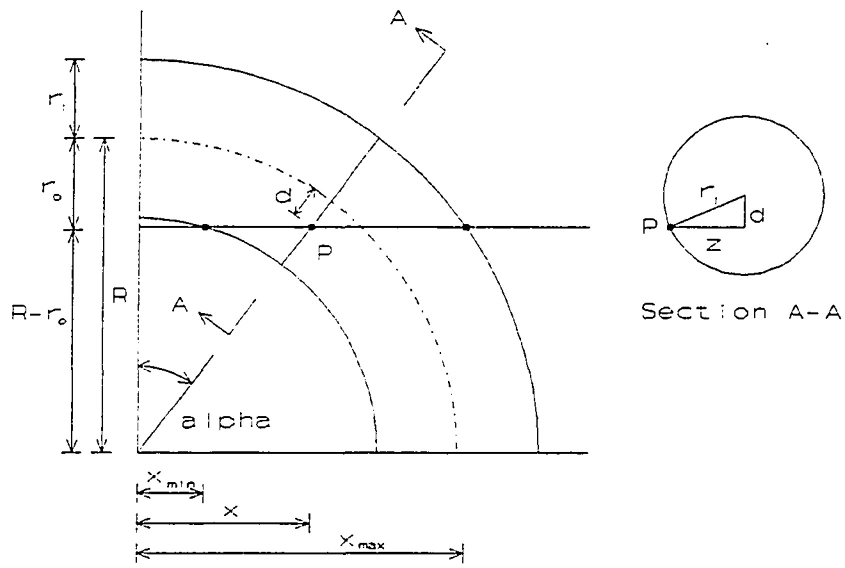


Figure 6.2: Intersection of a Torus and a Plane

$$d = R - \sqrt{(R-r_o)^2 + x^2} \tag{6.3}$$

6.2.2 Intersection of a Torus and a Cylinder

The shape of the tip of a BC&T type pitorifice can be approximated by a torus intersecting a cylinder. Figure 6.3 shows an x-y plane view of a quarter section of a torus with major and minor radii of R and r_i intersecting a cylinder with an axis tangential to that of the torus and with radius r_o. A solution of z in terms of x allows a face-on view to be constructed of the open area of a BC&T type pitorifice with an inside radius of r_i. By setting $r_i=r_o$ the solution for the outside surface can be found from the general solution.

The limits of x are still given by Equation 6.1. At any point P common to both the torus and the cylinder:

$$\begin{aligned} z^2 &= r_o^2 - (R-y)^2 \\ &= r_o^2 - R^2 + 2Ry - y^2 \end{aligned} \tag{6.4}$$

The problem is now reduced to solving for y. This is done by first noting that:

$$d^2 = r_i^2 - z^2 \tag{6.5}$$

and substituting Equation 6.4 for z^2 in Equation 6.5 to arrive at a relationship between y and d which does

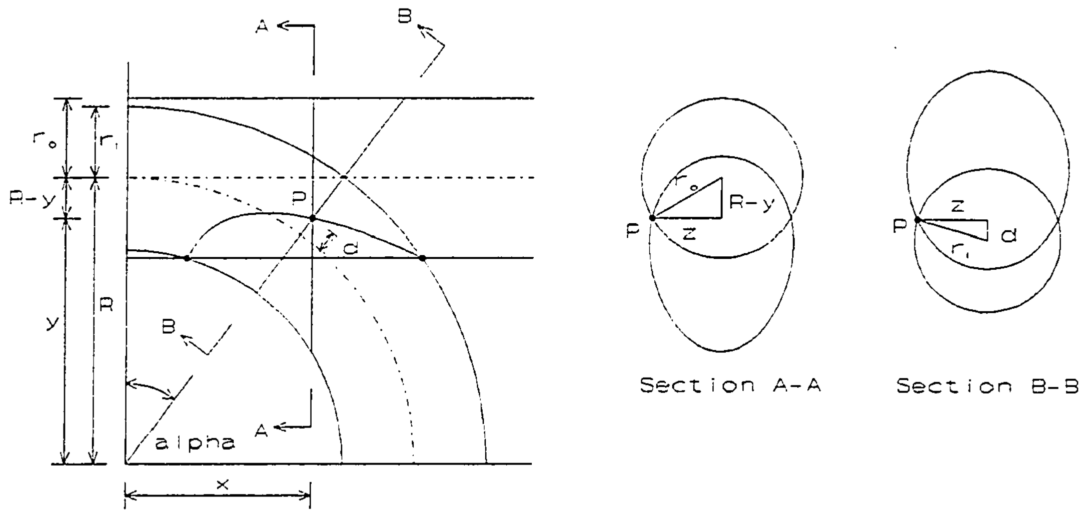


Figure 6.3: Intersection of a Torus and a Cylinder

not involve x or z:

$$\begin{aligned} d^2 &= y^2-2Ry+R^2-r_o^2+r_i^2 \\ &= y^2-2Ry-R^2-a \end{aligned} \tag{6.6}$$

$$\textit{Where} \quad a = (r_o^2-r_i^2)-2R^2$$

From Figure 6.3 it can be seen that:

$$(R+d)^2 = x^2+y^2 \tag{6.7}$$

$$R^2+2Rd+d^2 = x^2+y^2 \tag{6.8}$$

Rearranging Equation 6.8 and substituting Equation 6.6 for d^2 in Equation 6.8:

$$\begin{aligned} 2Rd &= x^2+y^2-R^2-y^2+2Ry+R^2+a \\ &= x^2+2Ry+a \end{aligned} \tag{6.9}$$

Squaring Equation 6.9 and substituting Equation 6.6 again for d^2:

$$4R^2(y^2-2Ry-R^2-a) = (x^2+2Ry+a)^2 \tag{6.10}$$

Expanding Equation 6.10, rearranging and collecting terms:

$$(4Rx^2+4Ra+8R^3)y = -x^4-2ax^2-a^2-4R^2a-4R^4 \tag{6.11}$$

Substituting back for a and solving for y:

$$y = \frac{\left[4R^2-2(r_o^2-r_i^2)\right]x^2-(r_o^2-r_i^2)^2-x^4}{4R(r_o^2-r_i^2+x^2)} \tag{6.12}$$

From Equation 6.4:

$$z = \sqrt{r_o^2-(R-y)^2} \tag{6.13}$$

For the case where $r_i=r_o$:

$$x_{max} = 2\sqrt{Rr_o} \tag{6.14}$$

$$y = \frac{4R^2-x^2}{4R} \tag{6.15}$$

$$z = \sqrt{r_o^2-\frac{x^4}{16R^2}} \tag{6.16}$$

The solution for the cross-section of the cylinder in Section B-B of Figure 6.3 is found by noting that if x and y are specified at a point P, then $y=x\tan\alpha$ and z can be determined from Equation 6.2. The limits are $y_{min}=R-r_o$ and $y_{max}=R+r_o$ and the cross-section is an ellipse.

A solution can also be found for d and z for $\alpha_{min}\leq\alpha\leq\alpha_{max}$ where α is the angle the x-y coordinates are rotated about the z-axis and R+d is the distance to the point of intersection along the displaced y-axis. From Figure 6.3 it is seen that:

$$\alpha_{min} = \arccos\left(\frac{R-r_o}{R-r_i}\right)$$

$$\alpha_{max} = \arccos\left(\frac{R-r_o}{R+r_i}\right) \tag{6.17}$$

From section A-A of Figure 6.3:

$$r_o^2 = (R-y)^2 + z^2 \tag{6.18}$$

From section B-B of Figure 6.3:

$$r_i^2 = d^2 + z^2 \tag{6.19}$$

Subtracting Equation 6.19 from Equation 6.18 to eliminate z^2:

$$r_o^2 - r_i^2 = (R-y)^2 - d^2 \tag{6.20}$$

Substituting for $y=(R+d)\cos\alpha$, expanding and collecting terms and solving for d:

$$\begin{aligned} d &= \frac{-b \pm \sqrt{b^2-4ac}}{2a} \\ \textit{where}\quad a &= 1-\cos^2\alpha \\ b &= 2R(\cos\alpha - \cos^2\alpha) \\ c &= r_o^2 - r_i^2 - R^2(1-2\cos\alpha+\cos^2\alpha) \end{aligned} \tag{6.21}$$

Solving for z from Equation 6.19:

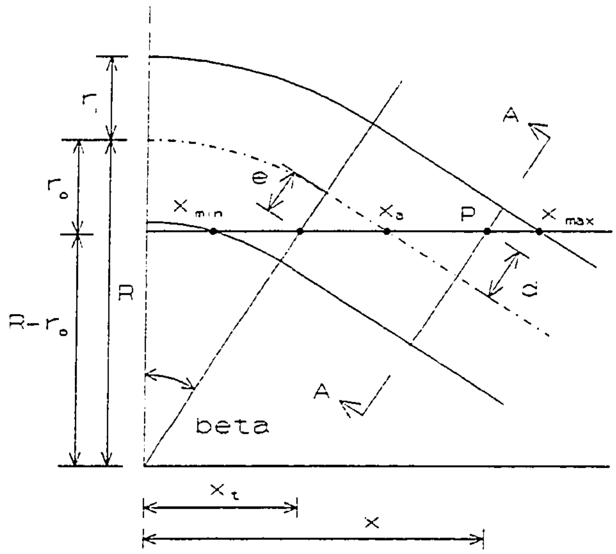
$$z = \pm\sqrt{r_i^2 - d^2} \tag{6.22}$$

Note that $d=-r_i$ at α_{min} and r_i at α_{max}.

6.2.3 Intersection of a Cylinder and a Plane

The shape of the tip of a IB&C type pitorifice can be approximated as a combination of a torus and a cylinder intersecting a plane. Figure 6.4 shows an x-y plane view of a section of a torus with major and minor radii of R and r_i joined to a cylinder with radius r_i or a view of a cylinder bent through an angle of β. The construction intersects a plane at $y=R-r_o$. A solution of z in terms of x allows a face-on view to be constructed of the open area of a IB&C type pitorifice with an inside radius of r_i. If the pitorifice is made by cutting flush along the unbent portion, as is normally the case, r_o represents the outside radius of the tube used for the pitorifice, otherwise r_o is the offset from the center line. By setting $r_i=r_o$ the solution for the outside surface can be found from the general solution.

Equation 6.1 is used for x_{min} and Equation 6.2 is used to find z for $x_{min}<x<x_t$ where x_t is given by:



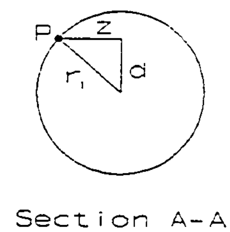


Figure 6.4: Intersection of a Cylinder and a Plane

$$x_t = (R - r_o)\tan\beta \tag{6.23}$$

From x_t to x_{max} the intersection of the cylinder with the plane $y=R-r_o$ can be described by an ellipse with major and minor dimensions of $2(x_{max}-x_a)$ and $2r_i$ and its center at $z=0$ and x_a where:

$$x_a = x_t + \frac{e}{\sin\beta} \tag{6.24}$$

and:

$$e = R - \frac{(R - r_o)}{\cos\beta} \tag{6.25}$$

Finally, x_{max} is found:

$$x_{max} = x_t + \frac{(e + r_i)}{\sin\beta} \tag{6.26}$$

Equation 6.2 can also be used for $x_a<x<x_{max}$ by finding d:

$$d = r_i - (x_{max} - x)\sin\beta \tag{6.27}$$

The projected area can be described as a rectangle with a height of $R\sin\beta$ and with half an ellipse on top as long as:

$$\beta \leq \arccos\left(\frac{R-r_o}{R}\right) \tag{6.28}$$

6.2.4 Intersection of a Cylinder and a Cylinder

The shape of the tip of an IBC&T type pitorifice can be approximated as a combination of a torus intersecting a plane and a cylinder intersecting a plane. Figure 6.5 shows an x-y plane view of a section of a torus with major and minor radii of R and r_i joined to a cylinder with radius r_i or simply a cylinder bent through an angle of β. The construction intersects a cylinder with radius r_o and an axis tangential to that of the torus or bent cylinder. A solution of z in terms of x allows a face-on view to be constructed of the open area of a IBC&T type pitorifice with an inside radius of r_i. By setting $r_i=r_o$ the solution for the outside surface can be found from the general solution.

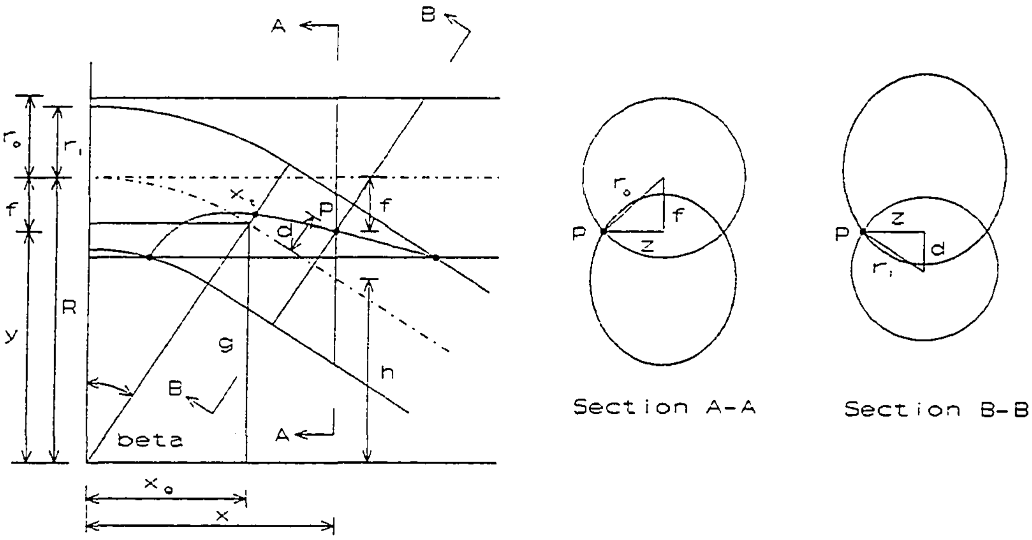


Figure 6.5: Intersection of a Cylinder and a Cylinder

Equation 6.1 is used to determine x_{min} and Equation 6.2 is used to find z for $x_{min}<x<x_t$ where x_t is given by:

$$x_t = (R+d)\sin\beta \tag{6.29}$$

and d is found using β in Equation 6.24. For $x_t<x<x_{max}$ first note in Figure 6.5 that:

$$x_g = R\sin\beta \qquad (6.30)$$
$$g = R\cos\beta$$

where x_g and g are the x and y coordinates where the axis of the torus transitions to the axis of the cylinder. By inspection it can be seen that:

$$\begin{aligned} x_{\max} &= x_g + \left[g - (R - r_o)\right]\cot\beta + \frac{r_i}{\sin\beta} \\ &= R\sin\beta + \left[R(\cos\beta - 1) + r_o\right] + \frac{r_i}{\sin\beta} \end{aligned} \qquad (6.31)$$

At any point P on the intersection:

$$f = R - y \qquad (6.32)$$

and:

$$y = h + \frac{d}{\cos\beta} \qquad (6.33)$$

where:

$$\begin{aligned} h &= g - (x - x_g)\tan\beta \\ &= R\cos\beta - (x - R\sin\beta)\tan\beta \end{aligned} \qquad (6.34)$$

From Sections A-A and B-B of Figure 6.5:

$$r_o^2 - f^2 - z^2 = 0 \qquad (6.35)$$

$$z^2 = r_i^2 - d^2 \qquad (6.36)$$

From Equation 6.33:

$$d = (y - h)\cos\beta \qquad (6.37)$$

Substituting Equation 6.37 into Equation 6.36 and then substituting Equations 6.36 and 6.32 into Equation 6.35 and collecting terms:

$$(1 - \cos^2\beta)y^2 + 2(h\cos^2\beta - R)y + (R^2 + r_i^2 - r_o^2 - h^2\cos^2\beta) = 0 \qquad (6.38)$$

This quadratic formula can be used to solve for y for any h:

$$y = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

$$\begin{aligned} where \quad a &= 1 - \cos^2\beta \\ b &= 2(h\cos^2\beta - R) \\ c &= R^2 + r_i^2 - r_o^2 - h^2\cos^2\beta \end{aligned} \tag{6.39}$$

and h in turn is found from x using Equation 6.34. For the special case where $r_i = r_o$ the intersection past x_t is seen as a straight line in the x-y plane and:

$$y = x_t \cot\beta + \frac{(R - r_o - x_t \cot\beta)x}{(x_{max} - x_t)} \tag{6.40}$$

Using Equations 6.32 and 6.35, z is found in terms of y:

$$z = \pm\sqrt{r_o^2 - (R - y)^2} \tag{6.41}$$

6.2.5 Projected Area of a Pitorifice in a Pipe

The total projected area of a pitorifice in a pipe is the sum of the areas of the top of the pitorifice and the exposed stem minus that part of the stem which is masked by the curvature of the pipe as shown in Figure 6.6. The area of the top of the pitorifice is either:

$$A_t = 2\int_0^{x_{max}} z\,dx \tag{6.42}$$

where z is defined in the previous subsections, or, in the case of pitorifices made from fittings, it is the projected area of the fitting. This area is shown in Figure 6.6 and it is given for the shapes and sizes used in Table 6.1.

Also shown in Figure 6.6 and given in Table 6.1 are the dimensions x_t, d_{o1}, d_{o2}, and h_c. For pitorifices made by bending tube, x_{max} is the length of the bent segment. For shape C x_{max} is the point halfway along the cut which is where the projected view changes from a rectangle to an ellipse. Because shape C was made by cutting the tube at 45° with $h_c = r_o$. For pitorifices made from fittings, x_{max} is the distance from the tip to the end of the fitting. The dimensions d_{o1} and d_{o2} correspond to the ODs of the first and second parts of the pitorifice assembly or the fitting cup and the tube. For the nominal 1-inch shapes and the nominal ¾-inch C shape there was no change in OD and $d_{o1} = d_{o2}$. The nominal ¾-inch

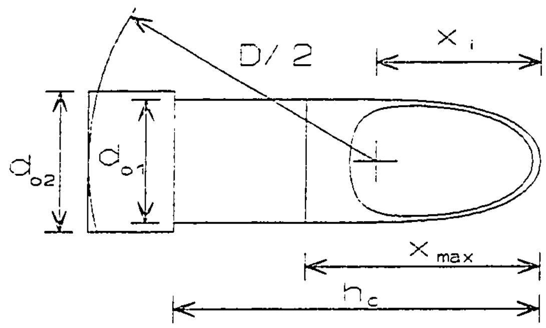


Figure 6.6: Projected Area of a Pitorifice in a Pipe

Table 6.1: Projected Areas and Dimensions of Pitorifices Used in Study

Nom. Size	Shape	A_t	x_{max}	d_{o1}	d_{o2}	h_c	R	β	x_t
3/4"	BC&T	1.05	1.47	0.810	0.875	2.15	1.34		
3/4"	IB&C	1.82	2.22	0.940	0.875	3.01	2.03	32°	
3/4"	IBC&T	1.18	1.84	0.750	0.875	2.10	1.87	32°	1.07
3/4"	C	0.30	0.44	0.875					
3/4"	C&B	0.27	0.90	0.750	0.875	2.00			
3/4"	L	1.68	1.83	0.975	0.875	1.83			
3/4"	LF	1.78	1.88	0.975	0.875	1.88			
3/4"	SQ	0.83	0.88	0.960	0.875	0.88			
3/4"	45L	1.90	2.22	0.970	0.875	2.22			
1"	BC&T	2.19	2.22	1.125			2.19		
1"	IB&C	2.67	2.73	1.125			2.69	32°	
1"	IBC&T	2.54	2.66	1.125			2.44	32°	1.40
1"	SQ	1.26	1.03	1.240	1.125	1.03			
1"	45L	3.00	2.72	1.240	1.125	2.72			
1/2"	45L	0.93	1.50	0.710	0.625	1.50			
3/8"	45L	0.65	1.21	0.580	0.500	1.21			

Dimensions given in inches and square inches.

BC&T, IBC&T and B&C shapes were obtained from pitorifices constructed of tube with a smaller OD than nominal ¾-inch diameter tube and the nominal ¾-inch IB&C shape was obtained from pitorifices with a larger OD than nominal ¾-inch diameter tube. For these cases the distance from the top of the pitorifice to the transition point is given as h_c. For pitorifices made from fittings, h_c is taken as x_{max}.

Also given in Table 6.1 are the bend radius R (measured to the center of the tube) for all the bent shapes, the bend angle β for all the incompletely bent shapes, and the dimension x_t for the IBC&T shapes. The bend radii were calculated by subtracting r_o from the bend radius of the outer edge of the tube which in turn was found to the nearest 0.25-inch using templates. The bend angle was measured with a protractor. The transition from a torus to a cylinder was calculated from the other information.

For the general case where $d_{o1} \neq d_{o2}$ a test must be made to determine if any tube with d_{o2} is exposed. If $D/2+x_i>h_c$:

$$A_T = A_t + d_{o1}(h_c - x_{\max}) + d_{o2}\left(\frac{D}{2} + x_i - h_c\right) - A_m \tag{6.43}$$

where D is the ID of the pipe, x_i is the insertion distance of the pitorifice past the center of the pipe (which makes its value negative if the pitorifice tip is not inserted all the way to the center), and A_m is the area masked by the curvature of the pipe. A_m is given by:

$$A_m \approx \frac{d_o}{6}\left(D - \sqrt{D^2 - d_o^2}\right) \tag{6.44}$$

where $d_o = d_{o2}$. If $D/2 + x_i < h_c$ Equation 6.43 reduces to:

$$A_T = A_t + d_{o1}\left(\frac{D}{2} + x_i - x_{\max}\right) - A_m \tag{6.45}$$

where A_m is calculated from Equation 6.44 using $d_o = d_{o1}$.

If the pitorifice stem OD changes and the pitorifice is inserted deep enough to expose that part of the stem, then A_a will have to be calculated using the different r_o, and A_s calculated in two parts.

CHAPTER 7: FULL AND REDUCED SCALE TEST RESULTS

Testing was done using nominal ¾ and 1-inch diameter Type K copper water tube, nominal ¾-inch threaded fittings, the pitorifices described in Chapter 6, and nominal 4 and 6-inch diameter Class 160 PVC pipe. Two approaches were considered for determining head losses in pitorifice pairs. The first was to vary the length of the service line and measure flows, then calculate the head loss that could be attributed to pitorifice losses by subtracting the other fitting and meter losses from the calculated pitorifice shut-off head. Service line lengths would then be changed to obtain different flows. The second approach, and the one that was used, was to measure the differential head across the pitorifices while throttling for different flow rates. This required the inclusion of calming lengths which had to be accounted for and the measuring of differential head. Service entrance fitting losses also had to be determined separately and could not be determined in combination with pitorifice losses. However, there was no requirement to subtract out fitting and meter losses and to change line lengths.

The following sections describe the test set-ups and present results.

7.1 SERVICE LINE FITTING LOSS TESTING

Because there were no reliable data on losses in fittings for laminar flow conditions and because losses in adjacent fittings cannot be considered additive, tests were conducted on nominal ¾-inch diameter Type K copper water tube with service loop entrance and tee-off plumbing standard for Fairbanks. These tests were among the first and the last conducted during the course of the study, and encompass a considerable evolution in the apparatus which was a result of both the experience gained and the acquisition of equipment and funding.

Water was initially supplied from a household service and wasted to the drain but water temperatures were found to fluctuate as a function of flow rates because of the long run of water supply piping through heated space. To correct this, supply water was passed through a coil of ½-inch OD copper tubing submersed in a water bath and lines were insulated to ensure uniform temperatures. Later, a small wet rotor style circulation pump was added to close the loop and that eliminated the need for the temperature stabilization tank and water wasting. A significant amount of heat was added to the water from the wet rotor pump so a magnetic drive pump was then substituted.

Initially, the head loss was measured with a water-air manometer and a water-oil dual-fluid manometer, flow was adjusted using a rotameter and throttling valve, flow was measured using timed fill of graduated cylinders, and temperatures were measured using thermocouples. When funding became available, a differential pressure transducer, a magnetic flow meter, and thermistors were substituted for the dual-fluid manometer, the rotameter, and the thermocouples, and data were automatically logged. Figure 7.1 shows a schematic of the final version of the head loss testing apparatus. Water was pumped from the insulated temperature stabilization tank and throttled by a downstream globe valve. After the globe valve, water flowed through 1-inch diameter vinyl hoses either upward through a ball valve to an adjustable constant head device or downward to the piping under testing. Ten-foot lengths of copper tube on either side of the test item served as calming lengths and contained piezometer taps. Water was returned to the tank through a magnetic flow meter, a ball valve, and an overflow. Thermistors were installed at the tank inlet and outlet. Flow rates were set by moving the constant head device up or down and adjusting the globe valve to ensure adequate flow. The ball valve on the line leading to the constant head device was closed at the beginning of a test run to allow very high flow rates for purging air. The ball valve before the overflow was occasionally closed to assist in purging air from manometer tubing.

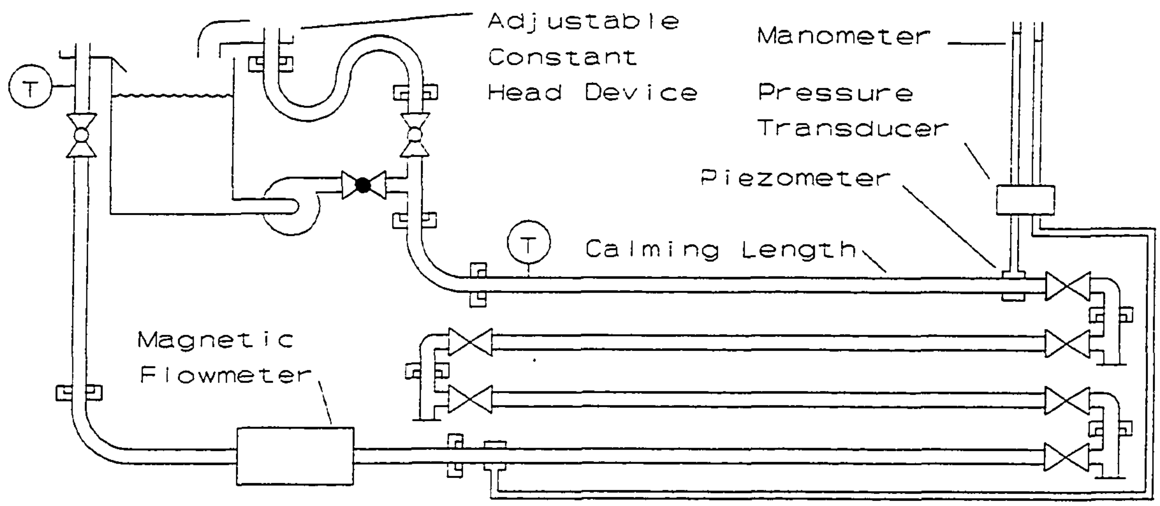


Figure 7.1: Service Line Fittings Head Loss Testing Apparatus

The adjustable constant head device was constructed by removing the saw and arm from a radial arm saw, leaving only the crank and adjustable elevating post mechanism. The saw table top was replaced and the tank, pump, and associated plumbing were securely mounted on the new table top. A shelf was installed below the table top and holes drilled in the table top and shelf to allow installation of the pressure transmission lines and the differential pressure transducer and manometer.

The constant head device consisted of a piece of nominal ¾-inch diameter copper tube installed in a modified 1-inch threaded brass tee which collected overflow and directed it through a pipe into the tank. The overflow at the tank consisted of a piece of nominal ¾-inch diameter copper tube which overflowed to a plastic basin which in turn allowed a stream of water to flow into the tank. Water from the overflow at the tank could be easily diverted and caught in a graduated cylinder for flow calibration of the meter. Both tubes overflowed evenly with the water surface appearing as a smooth dome.

After air was purged from all the lines, water was circulated until temperatures stabilized and then equipment was calibrated. Tests were started at the maximum flow rate to be used and the constant head device then lowered using the crank. One to five complete turns on the crank were made between measurement points depending on the step size desired. After waiting about one minute for flows to stabilize, data logging was initiated and 5-second means of five 1-second samples were logged. Tests were terminated at the maximum flow rate as a check on the first measurements logged. Values were noted by hand to assist in choosing appropriate log intervals because previously logged values were not available for immediate viewing, but logged values were used in the data reduction.

The nominal ¾-inch diameter Type K copper water tube service loops in Fairbanks typically terminate inside homes with a brass flare to male pipe thread (MPT) transition fitting, gate valve, pipe nipple, elbow, nipple, union, nipple, tee, nipple, gate valve, and MPT to flare fitting. Three plumbing terminations were used in the testing to provide a measurable head loss. They were made using Ford Meter Box Co. transition fittings, 125 psi gate valves, 2½-inch brass pipe nipples, and brass unions, elbows, and tees. The plumbing terminations were assembled as shown in Figure 7.1 with 10-foot lengths of nominal ¾-inch diameter Type K copper water tube between them to provide calming lengths. The total length of piping between the piezometers was 11.125 m [36.5 ft] of which 9.580 m [31.4 ft] was copper water tube and 1.545 m [5.1 ft] was fittings.

While nominal ¾-inch diameter Type K copper water tube has an ID of 0.745 inch, brass pipe has an ID of 0.824 inch. The brass pipe nipples had an interior lip with an ID of about 0.8 inch which was formed when the threads were cut. The ID of the transition fittings was 0.72 inch for a distance of about 1 inch, the ID of the gate valves was 0.71 inch for about 0.15 inch of length, widening to about 1-inch ID for about 0.6 inch, then constricting again to 0.71-inch ID for about 0.15 inch, the ID of the unions is about 0.92 inch for about 1 inch, and the ID of the cast elbows and tees was about 1.05 inches for about 1.5 inches along a bend. With only a few exceptions, therefore, most of the fitting length had an ID significantly larger than that of the copper water tube.

Fitting losses are determined by subtracting the losses that would have occurred in an equal length of straight pipe from the losses observed in the length of pipe with the fittings. The total length of line including fittings is then used in combination with fitting losses when calculating head loss. For this test the brass pipe nipples can be either considered as a fitting with the nominal ¾-inch diameter Type K copper water tube considered as the pipe or the nipples can be considered as the pipe itself. If the nipples are considered as fittings, the calculations are simpler but the results less generally applicable to different relative sizes of fittings and pipes. If the brass nipples are considered as the pipe, then pipe head loss must be calculated separately for both the brass pipe and the copper water tube, with the sum subtracted from the total head observed to yield the fitting coefficient. The fittings coefficient, K_{ftgs} can be expressed in terms of the velocity in either the nipples or the tube as given in Equation 4.23. Equation 4.23 can be combined with the Darcy-Weisbach equation (Equation 4.16) and solved for K_{ftgs}:

$$K_{ftgs} = \left[2gH - \frac{f_1 L_1 V_1^2}{D_1} - \frac{f_2 L_2 V_2^2}{D_2} \right] V_{ftgs}^{-2} \tag{7.1}$$

where H is the total observed head loss for the system, f is the Darcy friction factor, L is length, D is inside diameter, V is velocity, and the subscripts 1, 2, and ftgs refer to the two pipe sizes and fittings, respectively. The Darcy friction factor is given by Equations 4.17 and 4.18 for Reynolds numbers below 2,100 and above 4,000, respectively. It cannot be reliably calculated for intermediate Reynolds numbers.

Figure 7.2 shows the results of this later approach as a plot of the log of the total fitting loss coefficient, K_{ftgs}, for a single plumbing termination versus the log of the Reynolds number for the brass pipe for runs at three different temperatures. Equation 4.18 was used for Re>2,100 or log(Re)>3.3, so the graph cannot be considered valid beyond that point. However, all flows of interest in pitorifices will be in the laminar region. The data in the laminar region is fitted with a curve with K_{ftgs}=7446/Re−3.47 following the form of Equation 4.24.

A line representing the calculated loss for a 2-m [6.6-foot] length of 0.824-inch ID pipe is also shown in Figure 7.2. The close fit of this line with the data in the laminar region suggests it is reasonable to approximate the fitting losses as equivalent to an additional 2-m length of 0.824-inch ID pipe. This results in a total equivalent length of 2.5 m [8.2 ft] of pipe when the 0.5 m of true length is included. Equation 4.20 can be used to convert this to an equivalent length of 0.745-inch ID tube for a given head and mass flow rate:

$$L_2 = L_1 \left[\frac{d_2}{d_1} \right]^4 \tag{7.2}$$

where L and d are the length and ID of a pipe respectively and the subscripts refer to the two different sizes of pipe. Because the total loss for a plumbing termination is equivalent to 8.2 ft [2.5 m] of 0.824-inch ID pipe, the total equivalent loss of the pipe and fittings in terms of 0.745-inch ID tube is 5.6 ft [1.7 m] or an additional 4.0 ft [1.2 m] over the 1.6 foot [0.5 m] run length. Therefore, adding 4.0 ft [1.2 m] to the total length of 0.745-inch ID tube and 0.824-inch brass fittings yields the total equivalent length of the service loop in terms of 0.745-inch ID tube.

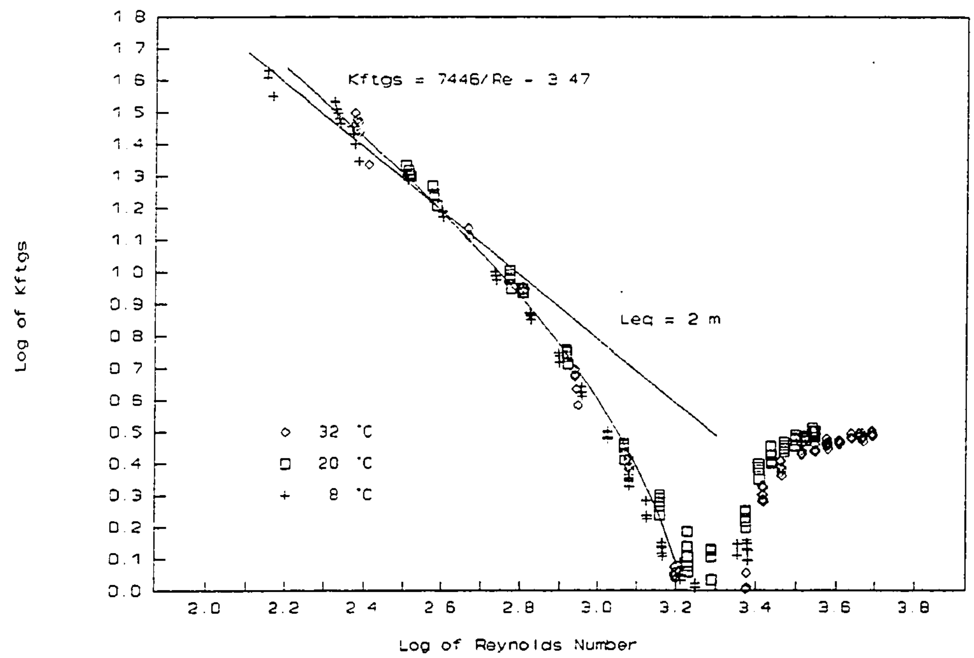


Figure 7.2: Total Fitting Resistance Coefficient Versus Reynolds Number

7.2 FULL SCALE PITORIFICE PERFORMANCE TESTING

The pitorifice performance testing encompasses loss in the main, head developed across the pitorifice pair, and loss in the pitorifice pair when water is flowing through the service line. The apparatus used and results obtained using various shapes of pitorifices are described in the following subsections. Only matching shapes were tested, consistent with the original design philosophy, so that future stocking requirements and the chance for installation errors would be kept to a minimum.

7.2.1 Test Loop Construction

It was originally planned to draw water from the return leg of a loop distribution main, divert it through a meter and a test length of pipe, and waste it to the clear well at the MUS water treatment plant. Pipe was ordered and a constant head tower built with this expectation, but the test location had to be relocated due to changes at the water plant made in the interim. The new location offered more space but made it more difficult to draw and waste water in large quantities, therefore a tank and pumps were added. The tank was a surplus chemical mixing vat from MUS and the pumps were surplus wet rotor Grundfos Series 6000 Duals Model UPSD 80-160 2 Hp pumps. The final arrangement is shown in Figure 7.3. Water was pumped from the tank using one or both pumps through a 3-inch butterfly valve and a 3-inch diameter hose to the inlet of the constant head tower. From there water flowed through a 4-inch butterfly valve through a magnetic flow meter and back to the tank. The line back to the tank could be 4 or 6-inch diameter and it contained piezometer rings and a 2-foot removable section which could either be a blank or which could contain pitorifice pairs. There was a second 4-inch butterfly valve on the main near the tank and a vent at the high point before the pipe discharged below the water level in the tank.

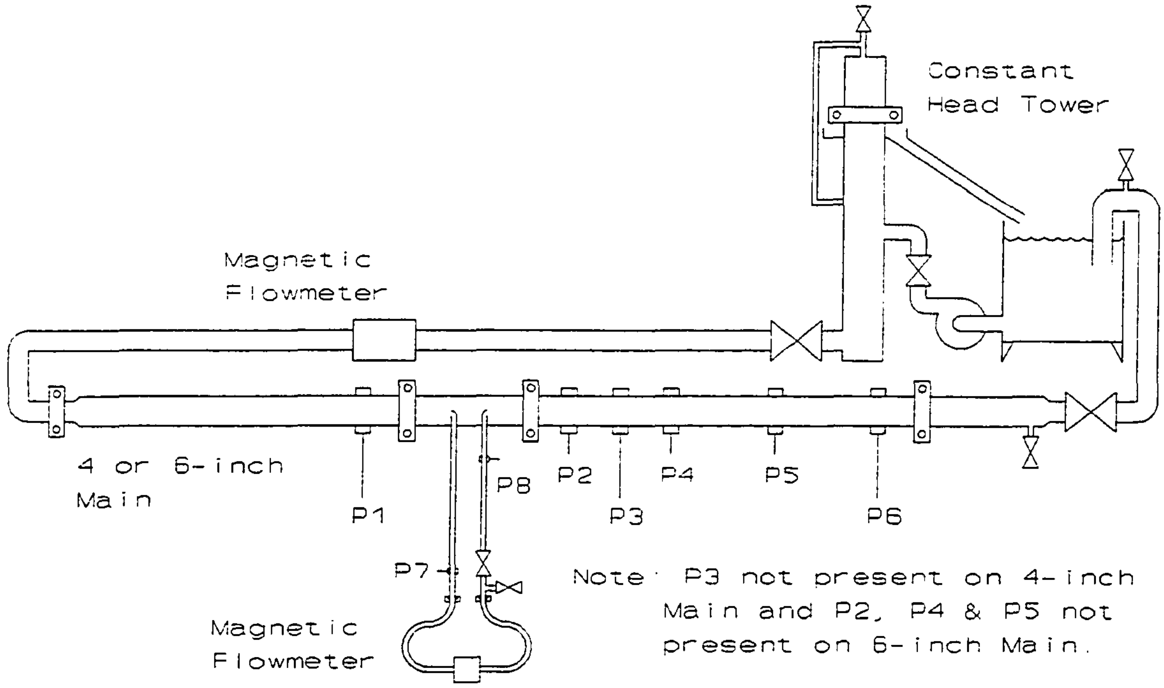


Figure 7.3: Pitorifice Test Loop Schematic

The tank was 36 inches in diameter and had ⅛-inch thick steel walls with a ⅛-inch thick epoxy lining. It had 4-inch high legs and the rim was 40 inches above the floor. A 1-inch drain valve was added at the bottom and a 4-inch flanged outlet was welded to the side near the bottom. An elbow was bolted to the flange and the dual pumps were bolted to the elbow. Because the pumps were operating under less than the recommended system pressure, the air purging vents for the pumps were modified so that water could be forced into the wet rotor annulus under pressure. This precaution did not prove necessary.

The constant head tower was intended to limit pressure and flow fluctuations. It consisted of an upright 6.5-foot piece of 8-inch diameter steel pipe with the bottom welded closed with a plate, a 3-inch flanged inlet midway up, a 4-inch flanged outlet at the bottom, and an overflow trap at the top made from 12-inch diameter pipe which drained back to the tank. As originally conceived, the tower was high enough to provide the necessary flow through 4-inch diameter test pipe sections, but would have to be elevated about 10 ft when using 6-inch diameter test pipe sections. The extension was to have been made with 12-inch diameter PVC pipe attached using a PVC coupling which had been machined to fit the steel pipe OD. As an alternative, a cap was fashioned as shown in Figure 7.3 with a sight glass and vent so that air could be trapped and the water level monitored and adjusted. There was no perceptible difference in the operation of the test apparatus when using the constant head tower as an overflow device or as a shock absorber. It is likely it could have been dispensed with altogether.

The PVC pipe was assembled with glue fittings, flanges, and Victaulic® grooveless couplings. Flanges were used at the butterfly valves (which were all wafer type) and at the magnetic flowmeter. Grooveless couplings were used to allow interchanging the return test section length with either 4-inch or 6-inch diameter pipe and to allow the 2-foot length with pitorifices to be replaced within the return length. The Victaulic couplings were not intended for use on plastic pipe and there was a risk of the pipe being cracked, but they worked well. An alternative coupling method which used PVC couplings sealed with tape or rubber strips was also tried but it did not offer any alignment or assembly advantages so it was not used. The PVC couplings had to be machined out so that they could be slid completely back from a joint allowing a section to be removed without having to pull the pipe ends away from each other and these were more troublesome to slide and reseal than the Victaulic couplings were to unbolt.

Pipe was supported off the floor with plywood saddles that were shimmed to allow the test section to drain. This minimized spilling when sections of pipe were replaced. The drain also served for wasting water. During operation, the water would warm about 1.5 C°/hr so water was bled to waste and cold make-up water added. An automatic fill valve was added and the bleed rate was manually adjusted. Most testing was done at 21°C ±1 C° for convenience, but some tests were done at 11°C, the temperature of the supply water, to investigate effects of temperature.

The entire replaceable test section was 36.5 ft [11.12 m] long. It started at a Victaulic coupling to the 4-inch diameter pipe, and consisted of a 21.60-foot [6.89-m] calming length, a Victaulic coupling, a 2.00-foot [0.61-m] removable section which could be used for pitorifices, a Victaulic coupling, a 7.42-foot [2.26-m] section, a Victaulic coupling, and a final 4.46-foot [1.36-m] section which terminated in a 4-inch flange which connected to the wafer style butterfly valve. The initial and final sections included 6×4-inch reducers in the case of the 6-inch diameter test section. Five piezometers were installed on the 4-inch diameter test section and three on the 6-inch diameter test section at the locations shown in Figure 7.3 as P1 to P6. P1 was 1 foot [0.30 m] upstream of the removable 2-foot section and P6 was 9.42 ft [2.87 m] downstream from P1. Piezometers were installed at intervals of one-fifth the distance between P1 and P6, skipping the first location, on the 4-inch diameter test section and at one-half way between P1 and P6 on the 6-inch diameter section. This resulted in 1.88-foot [0.574-m] intervals at locations P2, P4, and P5 on the 4-inch diameter test section and 4.71 ft [1.435 m] intervals at locations P3 and P6 on the 6-inch diameter test section.

Pitorifices were inserted into the main using modified compression fittings so that the insertion depth and the orientation could be easily varied. The compression fittings accommodated nominal ¾ or 1-inch diameter copper water tube. The upstream pitorifice was attached to a length of tube which served as a calming length. A piezometer (P7 in Figure 7.3) was installed on the end of the calming length, then 1-inch ID vinyl hoses were use to connect a magnetic flowmeter to a tee. The tee had ball valves on both discharges which allowed air bleeding, calibration of the flowmeter, and throttling of the flow. A second length of tube served as another calming length and it was followed by another piezometer (P8 in Figure 7.3) and the downstream pitorifice. The pitorifice tubes were labelled to show shape and direction and marks were scratched on the circumferences 222 and 255 mm [0.73 and 0.84 ft] from the tip of the nominal ¾ and 1-inch pitorifices respectively. The marks were used as a reference point when positioning pitorifices in the main. The distances from the mark on the upstream pitorifice to piezometer P7 were 3.133 and 3.153 m [10.28 and 10.34 ft] and from P8 to the mark on the downstream pitorifice were 65 and 78 mm [0.213 and 0.256 ft] for the ¾ and 1-inch pitorifices respectively so total run lengths were 3.642 and 3.741 m [1.95 and 12.27 ft] respectively. These distances were used in calculating the head loss due to piping which was subtracted from the observed head loss to yield pitorifice pair head loss.

Connections on the pitorifice line were made using sweat fit couplings with one side left unsoldered. Electrical splice tape was used to make the unsoldered joint water tight.

7.2.2 Head Loss in the Main

Figure 7.4 shows the head loss in the 6-inch diameter test section between piezometers P1 and P3 and between piezometers P3 and P6. Each section was 4.71 ft [1.435 m] in length but the first contained a blank with joints. Both sets of data were obtained using the same transducer. The close agreement between the two sets of data show that the joints in the test section do not contribute significantly to head loss in the main and that their presence can be ignored. Each datum point is actually the mean of 50 values taken at one-second intervals. The data were closely fitted by assuming a hydraulic roughness of $\epsilon=0.07$ mm [0.00023 ft] and Equations 4.16 and 4.18. Moody (1944) gave values of $\epsilon=0.0006$, 0.00015, and 0.000005 ft [0.18, 0.046, and 0.0015 mm] for galvanized iron, commercial steel, and drawn tubing, respectively, so the computed ϵ for the PVC pipe seems reasonable. Moody noted that there was a wide variation in roughness for a given pipe material and wrote that the scale of ϵ was not based on measurements made directly on the pipe (i.e. on ϵ_0, the absolute geometric roughness) but was chosen based on experiments with artificially roughened pipe done by Nikuradse (1933). Moody noted that for commercial steel the variation in the value calculated for f and the true value could be 10 percent. To achieve this variation in the value of f=0.02 calculated for the 6-inch diameter PVC pipe at the mid-range Reynolds number of 100,000 requires $0<\epsilon<0.15$ mm. Idelchik (1986) gives a range of $\epsilon=0.07$ to 0.15 mm for galvanized iron citing a 1954 study but he does not give any values for plastic pipe. Lamont (1981) gives $\epsilon=0.03$ mm for 4.5 to 12-inch diameter wavy PVC at 3 fps. Using $\epsilon=0.03$ mm results in the calculated head being 94 percent what it would be for $\epsilon=0.07$ mm.

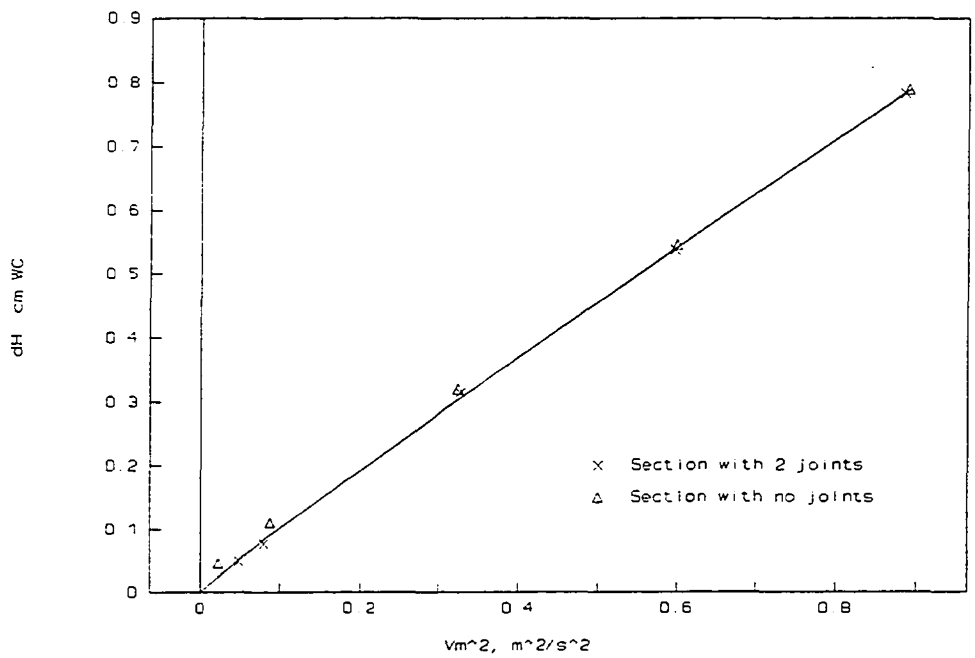


Figure 7.4: Head Loss in 1.435-m Lengths of 6-inch PVC Pipe

Although ϵ cannot be determined directly from inspection of the pipe surface, a roughness or waviness greater than that of a drawn tubing is indicated by the higher value of ϵ. The inner surface of the PVC pipe was smooth but had a clearly visible annular waviness. A strip of the pipe was cut out and secured to the bed of a milling machine under a dial gauge. The bed was advanced in 2 mm steps and the dial gauge read to the nearest 0.01 mm division mark. The data were fit using linear regression to allow correction for the slope of the clamped strip and the variation from that line is shown in Figure 7.5. The waves can be seen to have a length of about 15 mm and an amplitude of about 0.05 mm. Tietjens (1934) differentiated between roughness and waviness in pipe walls and noted that the friction factor for wavy walls was not as independent of Reynolds number as for rough walls but this was presumably for complete turbulence whereas in this study the Reynolds numbers were well within the transition zone.

The ID of the pipe was found to average 6.083 inches [154.5 mm] and this value was used although the standard maximum ID for this pipe is 6.101 inches [155 mm]. This variation is insignificant as far as head loss is concerned.

Because the head loss for a given velocity was higher in the 4-inch diameter pipe than in the 6-inch diameter pipe, head loss measurements over shorter sections could be measured with suitable accuracy. Figure 7.6 shows the head loss in the 4-inch diameter test section measured between piezometer P1 and piezometers P2, P4, P5, and P6 corresponding to lengths of 1.15, 1.72, 2.30 and 2.87 m [3.77, 5.64, 7.55, and 9.42 ft] respectively. Each datum point is actually the mean of 5 values taken at one-second

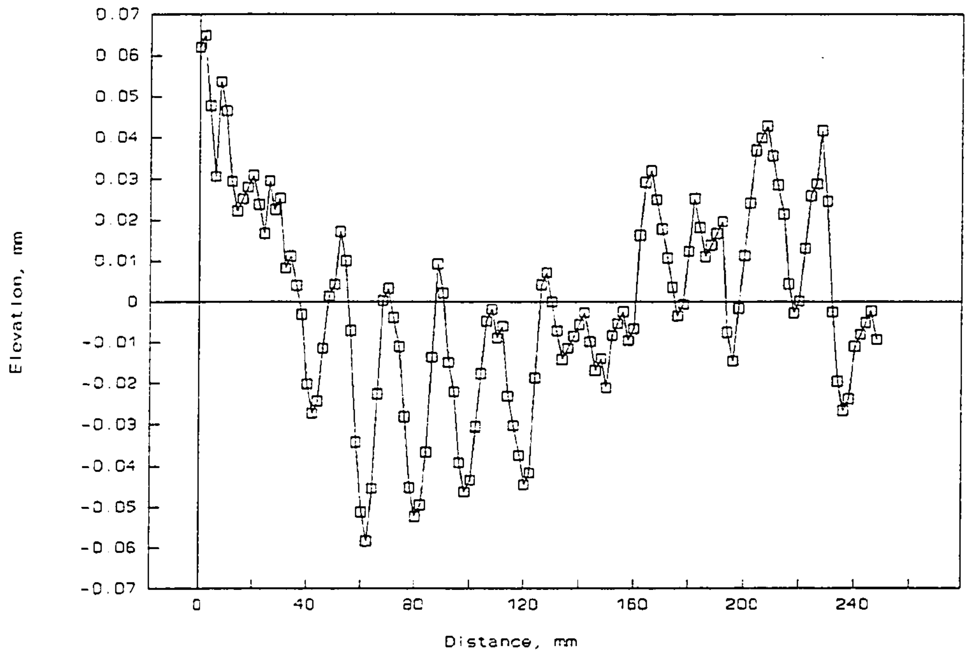


Figure 7.5: Annular Waviness Inside 6-inch PVC Pipe

intervals. The test was started with a velocity of about 0.9 m/s and the tubing to each of the appropriate piezometers was opened briefly and values logged before throttling the flow to a lower value. The test was terminated with a velocity of about 0.9 m/s so that there was a check on repeatability. The lines plotted were calculated using a hydraulic roughness of ϵ=0.08 mm. The close agreement between the points and lines for the different lengths up to 2.30 m indicates that, like the case for the 6-inch diameter pipe, the joint between piezometers P1 and P2 does not add significant head loss. No explanation was found for the lower than expected head loss values for the 2.87 m length. There may have been some effect from the downstream pipe which caused a local rise in the static pressure at P6. No change in pipe diameter or the apparent roughness was seen. The test was repeated twice, once at 21°C and once at 11°C, and the same effect was seen. Because of this effect, piezometer P5 and not P6 was used in testing head loss from pitorifices in 4-inch diameter pipe.

To ensure the downstream piezometer was sufficiently far removed from pitorifices being tested to register all downstream head losses resulting from the pitorifices, the previous test was repeated with a pitorifice pair inserted in the 2-foot section. Figure 7.7 shows the head loss from P2 to P4, P4 to P5, and P5 to P6 labeled as the First, Second, and Third Sections respectively. Also plotted is a line calculated using a hydraulic roughness of ϵ=0.08 mm and a length of 0.57 m [1.87 ft]. The close agreement between the first two sections and the plotted line indicates that the head loss from a pitorifice pair can be measured between piezometers P1 and P2, if desired.

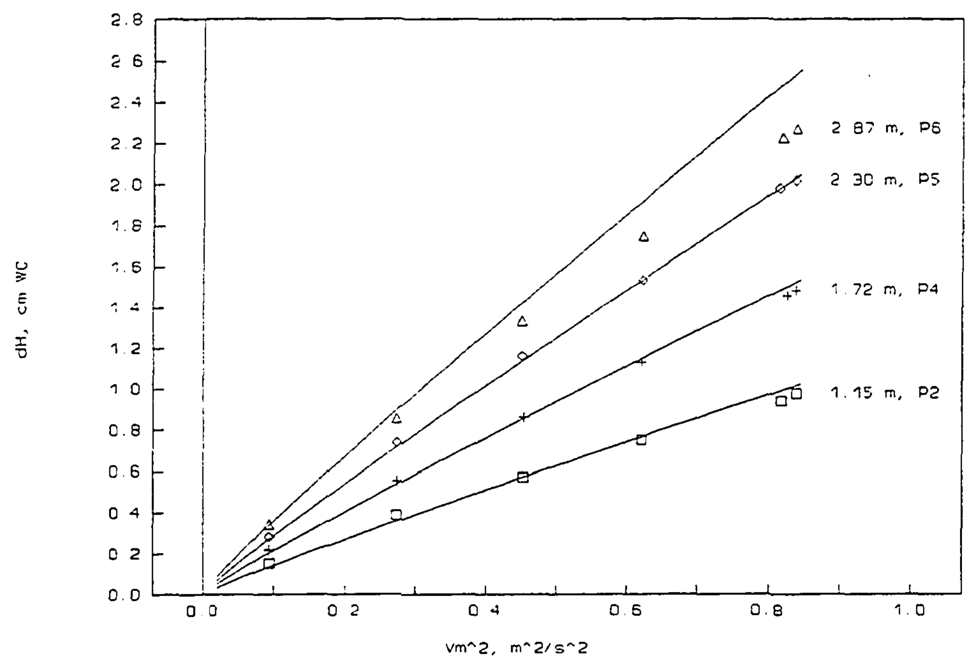


Figure 7.6: Head Loss in Cumulative Lengths of 4-inch PVC Pipe

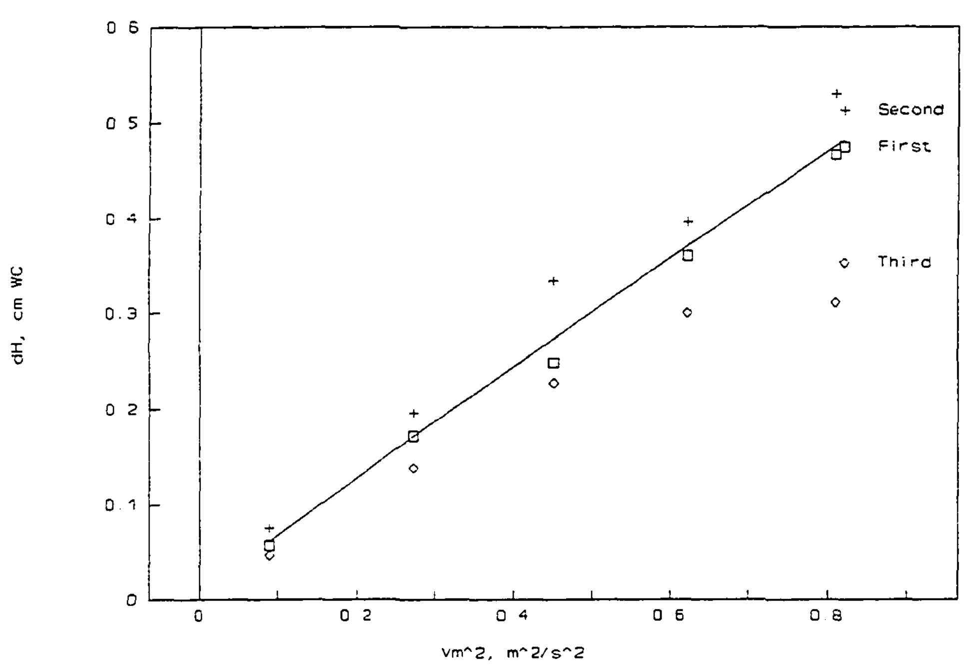


Figure 7.7: Head Loss in 0.57-m Lengths of 4-inch PVC Pipe Downstream of Pitorifices

7.2.3 Dye Observations in the Service Line

To ensure that the calming length was adequate and to observe transition from turbulence to laminar flow, a section of 0.75-inch ID clear acrylic plastic pipe was installed after an upstream BC&T pitorifice with its tip inserted to the center of a 6-inch diameter test pipe section. (See Figure 6.1 and Table 6.1 for drawings and dimensions of the pitorifice shapes.) A $^1/_{16}$-inch diameter hole was drilled in the acrylic pipe a distance of 0.35 m from the tip of the pitorifice and covered with rubber tape. After purging air from the apparatus, a flow of about 0.7 m/s was maintained in the 6-inch diameter pipe and flow was throttled through the pitorifice loop tubing. Food coloring was injected in the hole and laminar flow was observed to transition to irregularly turbulent between Reynolds numbers of 1,900 and 2,200. Even at a Reynolds number of 3,000, the highest obtained, the dye stream was not well mixed although it was definitely irregular. Below 1,900 the dye stream was very even and could be observed to spread out in a parabolic shape, indicating that less than 0.35 m was needed to calm the turbulent water entering the pitorifice and that no turbulent slugs were present.

7.2.4 Sensitivity to Orientation, Separation, and Temperature

To determine the effect of orientation on pitorifice performance, ¾-inch C&B pitorifices were inserted with their tips at the center of the 4-inch diameter test section and the pitorifice performance coefficient, K_{po}, was calculated by rearranging Equation 3.3:

$$K_{po} = \frac{2gH_{po}}{V_m^2} \tag{7.3}$$

where H_{po} is the measured head differential between the pitorifices with no flow through them (piezometers P7 to P8), and V_m is the mean or bulk velocity in the main. The mean of 15 values of K_{po} calculated for velocities of about 0.7 m/s were 2.25, 2.20, and 1.87 for the cases where the pitorifices were oriented directly up and downstream, rotated 22½° clockwise, and rotated 45° clockwise respectively. The close agreement between the first two values shows that pitorifices are not sensitive to minor variations in orientation.

To determine the effect of separation on pitorifice performance, ¾-inch BC&T pitorifices were inserted with their tips at the center of the 4-inch diameter test section. The heads across piezometers P1-P7, P7-P8, and P1-P2, P1-P4, or P1-P5 were logged at four or five different velocities in the main for 6-inch and 16-inch pitorifice separations. Velocities were chosen to take full advantage of the range of the differential pressure gauge and for good data distribution. Data are plotted in Figure 7.8 as a function of V_m^2 so that the slope of a line determined by linear regression yields values for the K_{po}s, K_{up}s, and C_fs as shown. The head loss that could be attributed to a 10-inch [0.254-m] diameter section of pipe with a hydraulic roughness of 0.08 mm is also shown. Head loss differences observed at a given V_m can yield a better understanding of the relative effect of the upstream and downstream pitorifices; this and the effect of separation will be discussed later. At V_m=0.77 m/s (V_m^2=0.6 m^2/s^2 on the x-axis in Figure 7.8) the head developed across the pitorifices is 7.2 mm WC with 6 inches separation and 7.7 mm WC with 16 inches separation for an increase of 5 mm WC, but the extra 10 inches of main can only be contributing 1.5 mm WC of this. Part of the difference may be due to a small increase in the head loss due to the pitorifices. More likely reasons for the difference are that a larger part of the head loss from the upstream pitorifice is developed or the pressure drag of the downstream pitorifice has a greater magnitude when the separation is greater. This suggests that all of the head loss attributable to the presence of the upstream pitorifice is not developed in the first 6 inches of separation. Presumably, decreasing the distance between the pitorifice will result in a further loss in performance, but this was not measured because current practice is to use saddles for tapping a main which result in a minimum separation of around 6 inches.

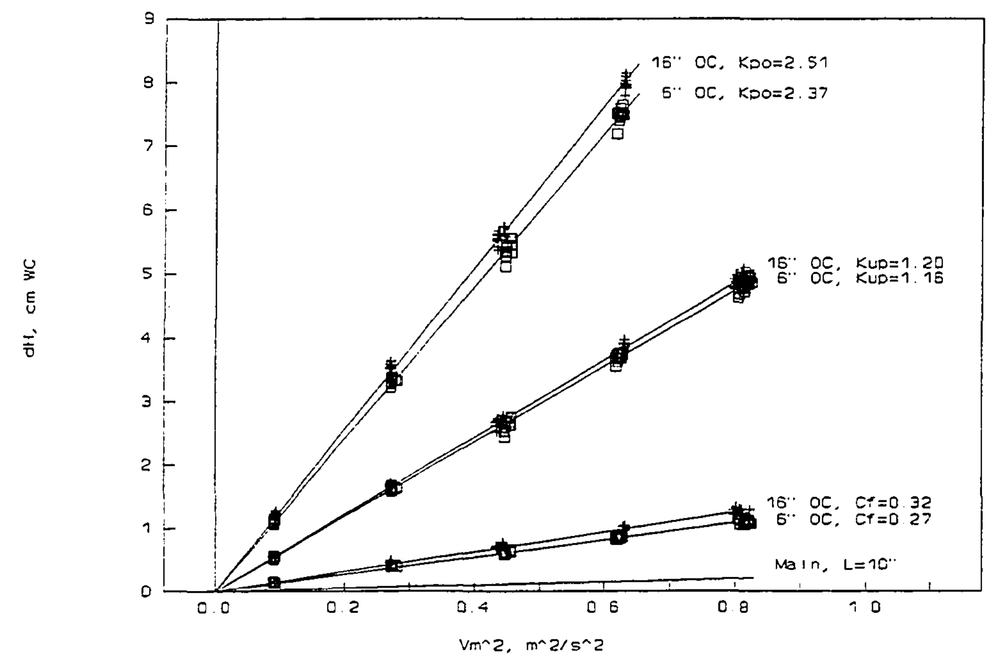


Figure 7.8: Effects of Separation of ¾-inch BC&T Pitorifices in 4-inch Pipe

To determine the effect of temperature on the overall performance of the pitorifice pair and on the upstream facing pitorifice alone, 1-inch IBC&T pitorifices were inserted with their tips at the center of the 6-inch diameter test section and 1.5 and 3.0 cm on either side of the center. Flows in the main were adjusted from 0.9 m/s to 0.3 m/s in four steps calculated to give equal spacing for V_m^2. The temperature, position, flow rate, and the differential pressures across piezometers P1 and P7 and piezometers P7 and

P8 were logged. Position was indicated by manually adjusting a resistor to result in a logged value of between −3 and +3 to represent insertion relative to the center. Data logging was started with the pitorifice tips inserted to the center of the pipe and with the flow in the pipe at 0.9 m/s. The data logger was activated for a period of 30 seconds after conditions stabilized. The data logger was programmed to sample every one second and store a mean value every five seconds. When the data logger was activated it would store a raw value, then the mean of anywhere from one to five values when its clock hit a five-second mark, then mean values for sets of five readings every five seconds afterward. Therefore the 30+ second interval resulted in seven or more logged values, the first two of which had to be discarded. After deactivating the data logger, the flow was adjusted to a lower value for another log sequence. When the minimum flow was reached, the flow was reset to the high value and a second set of values were logged. Then the pitorifices were adjusted in or out of the main and the entire process repeated. After all positions were logged the pitorifices were repositioned with their tips in the center and a second set of logs for the various flows was taken. This redundancy served as a check on instrument drift, errors, and reproducibility.

Water was maintained at 10°C for the first set of tests and at 21°C for the second set. Water temperature was manually adjusted to within ±1 C° by bleeding while introducing fresh water. The differential pressures across the pitorifices are plotted in Figures 7.9 and 7.10 as described for Figure 7.8. The values shown for the K_{po}s indicate no appreciable effect of temperature on performance. This is expected because most of the differential pressure is due to velocity head and downstream pressure drag effects which are not sensitive to viscosity.

Piezometer P1 was 1.75 ft [0.533 m] upstream of the upstream facing pitorifice. The head loss for that length of pipe was calculated and subtracted from the measured differential and that value was plotted against V_m^2 in Figures 7.11 and 7.12. Again, as expected, there was no appreciable effect of temperature indicated.

7.2.5 Pitorifice Performance

The primary objective of this phase of the study was to get performance data on existing pitorifice shapes. Secondary to this was the objective of analyzing and comparing performances so that the best present shape could be identified and possibly a better shape designed.

The performance of ¾ and 1-inch existing and experimental shapes for pitorifices described in Chapter 6 were determined for various insertion depths in both the 4 and 6-inch diameter test sections using the same method described in Subsection 7.2.4. Because nominal pitorifice and main sizes can vary significantly, it is important to refer to the exact sizes when applying the data developed in the study to other situations. Most testing was done on ¾-inch pitorifices in the 6-inch diameter pipe because the

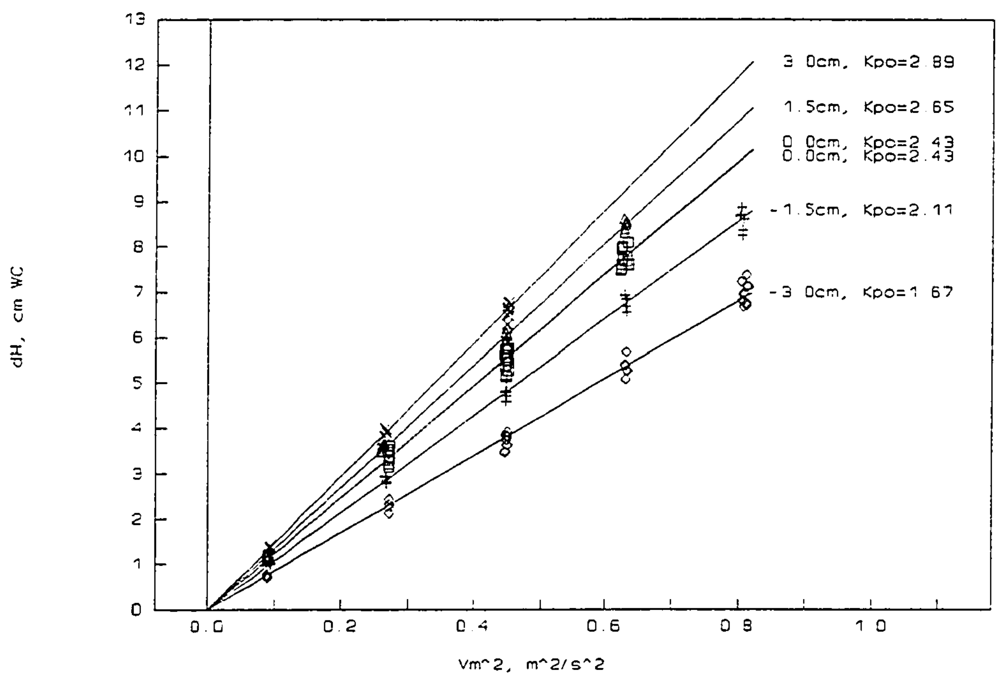


Figure 7.9: K_{po}s for 1-inch IBC&T Pitorifices in 6-inch Pipe at 10°C

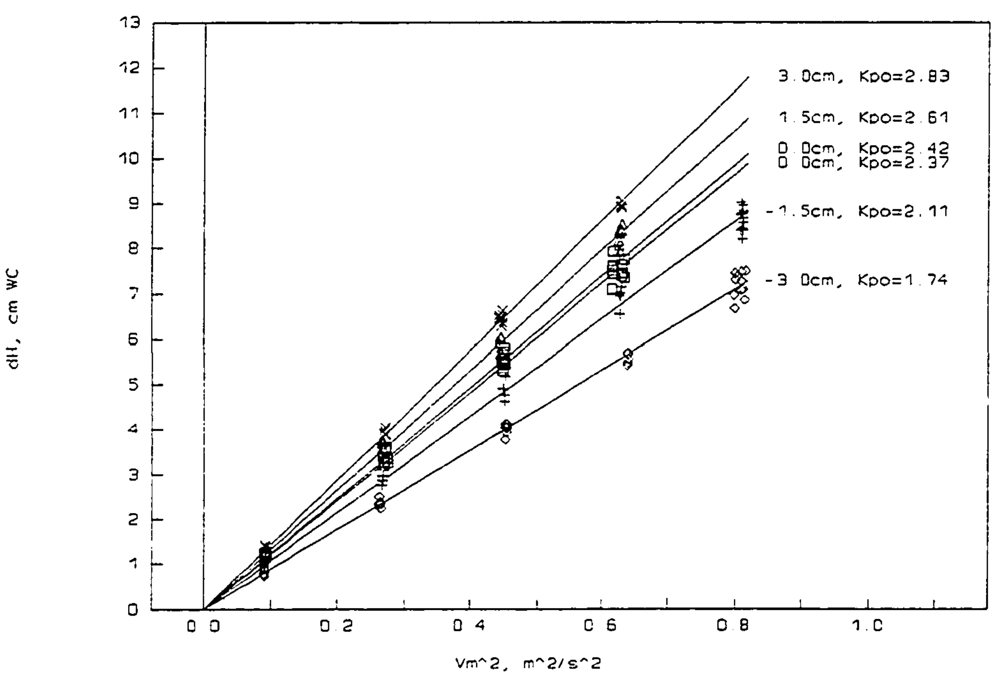


Figure 7.10: K_{po}s for 1-inch IBC&T Pitorifices in 6-inch Pipe at 21°C

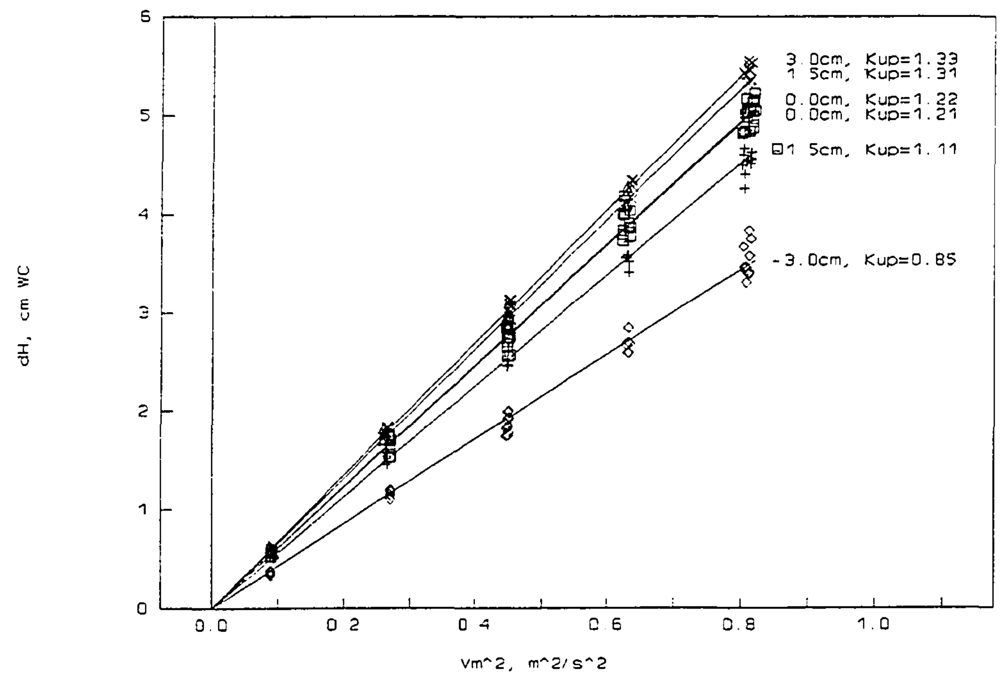


Figure 7.11: K_{up}s for 1-inch IBC&T Pitorifices in 6-inch Pipe at 10°C

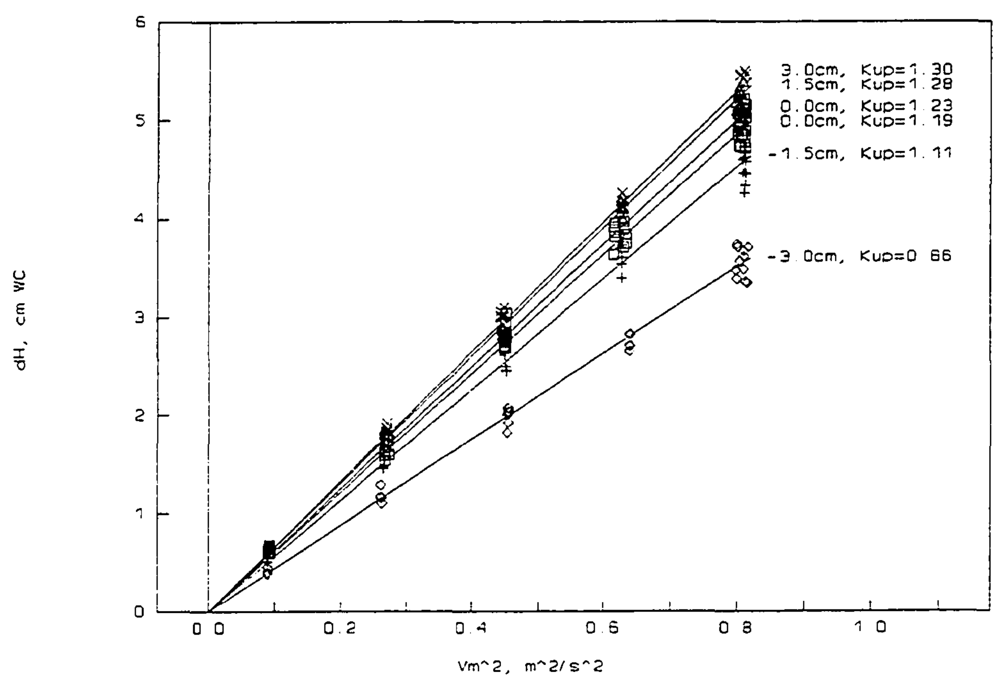


Figure 7.12: K_{up}s for 1-inch IBC&T Pitorifices in 6-inch Pipe at 21°C

blockage effect was less and the range of movement was greater. The larger size pitorifices, particularly those that were not turned in a lathe or that were made from fittings, often could not be positioned very far back from the center of the pipe. The plots of the data for the 6-inch diameter sections are contained in Appendix B and the results for all the tests are tabulated in Tables 7.1 and 7.2. The original shape introduced in 1953 was the BC&T, so it was chosen as a standard and used in all combinations. All eight of the ¾-inch pitorifices in 6-inch diameter pipe are roughly comparable in performance with $2.0<K_{po}<2.2$ and $1.2<K_{up}<1.4$ when the pitorifice tip is inserted to the center of the pipe, with the exception of the C shape pitorifice which has $K_{po}=1.6$ and $K_{up}=0.6$.

Table 7.1: K_{po}s for Pitorifices at Different Insertion Depths

Nom. Main	Nom. PO	Type PO	K_{po} for insertion of (cm): -3.0	-1.5	0	0	Avg.	1.5	3.0
6"	3/4"	BC&T	1.77	1.95	2.16	2.17	2.17	2.31	2.38
6"	3/4"	IB&C	1.88	2.08	2.24	2.31	2.28	2.38	2.58
6"	3/4"	IBC&T	1.84	1.97	2.15	2.17	2.16	2.37	2.45
6"	3/4"	C	1.31	1.56	1.59	1.61	1.60	1.66	1.68
6"	3/4"	C&B	1.68	1.91	2.09	2.10	2.10	2.21	2.41
6"	**3/4"**	**L**	**1.68**	**1.88**	**2.04**	**2.05**	**2.05**	**2.17**	**2.25**
6"	3/4"	LF	1.72	1.93	2.14	2.15	2.15	2.29	2.35
6"	3/4"	45L		2.03	2.15	2.18	2.17	2.36	2.47
6"	1"	BC&T	1.84	2.14	2.38	2.40	2.39	2.69	2.68
6"	1"	IB&C		2.14	2.44	2.46	2.45	2.65	2.80
6"	1"	IBC&T	1.74	2.11	2.37	2.42	2.40	2.61	2.83
6"	1"	45L			2.37		2.37	2.50	2.67
4"	3/4"	BC&T			2.37		2.38		
4"	3/4"	SQ			2.23	2.24	2.24		2.26
4"	3/4"	45L			2.43	2.46	2.45		2.39
4"	1"	BC&T			2.58	2.58	2.58		
4"	1"	IB&C			2.63	2.66	2.65		
4"	1"	SQ			2.08	2.11	2.10		

While the K_{po}s for all pitorifice shapes increase with insertion depth, the K_{up}s for four shapes decrease with insertion past the pipe center. The decrease in K_{up}s is expected because the upstream pitorifice will behave like a pitot tube. The flow velocity distribution in a pipe is given by Equation 4.4. The head measured by a pitot tube varies with the square of the velocity at the opening so Equation 4.4 can be used to find the K_{up} that would be given by an ideal pitot tube using k=1.20 and n=8 to result in:

$$K_{up,ideal} = 1.44\left[1-\frac{2r}{D}\right]^{0.25} \tag{7.4}$$

where D is the inside pipe diameter and r is the distance of the center of the pitot tube opening from the center of the pipe. The geometric center of the opening in the L pitorifice is 1.24 cm from the tip. Using this offset K_{up}s for r=4.24, 2.74, 1.24, 0.26, and 1.76 cm for insertions of $x_i=-3.0$, -1.5, 0, 1.5, and 3.0 cm are 1.18, 1.29, 1.38, 1.43, and 1.35 cm, respectively. These are in excellent agreement with the values listed in Table 7.2 for the L pitorifice. The geometric centers of the projected openings of the other

Table 7.2: K_{up}s for Pitorifices at Different Insertion Depths

Nom. Main	Nom. PO	Type PO	K_{up} for insertion of (cm): -3.0	-1.5	0	0	Avg.	1.5	3.0
6"	3/4"	BC&T	1.09	1.20	1.27	1.27	1.27	1.27	1.27
6"	3/4"	IB&C	1.08	1.23	1.32	1.35	1.34	1.38	1.38
6"	3/4"	IBC&T	1.10	1.19	1.26	1.27	1.27	1.28	1.31
6"	3/4"	C	0.56	0.61	0.57	0.57	0.57	0.55	0.48
6"	3/4"	C&B	1.03	1.15	1.22	1.23	1.23	1.24	1.25
6"	**3/4"**	**L**	**1.19**	**1.30**	**1.37**	**1.37**	**1.37**	**1.39**	**1.35**
6"	3/4"	LF	1.15	1.28	1.36	1.37	1.37	1.37	1.39
6"	3/4"	45L		1.15	1.21	1.23	1.22	1.30	1.30
6"	1"	BC&T	1.03	1.16	1.23	1.24	1.24	1.30	1.30
6"	1"	IB&C		1.18	1.29	1.30	1.30	1.35	1.38
6"	1"	IBC&T	0.86	1.11	1.19	1.23	1.21	1.28	1.30
6"	1"	45L			1.23		1.23	1.26	1.30
4"	3/4"	BC&T			1.16		1.16		
4"	3/4"	SQ			1.15	1.16	1.16	1.16	1.15
4"	3/4"	45L			1.17	1.19	1.18		1.07
4"	1"	BC&T			1.06	1.07	1.07		
4"	1"	IB&C			1.08	1.08	1.08		
4"	1"	SQ			1.06	1.08	1.07		

¾-inch pitorifices are all greater than 1.24 cm from the tips so they can be expected to lag these values a small amount, plus the others can all be expected to exhibit more of an effect from probe blockage. The three-dimensional shapes of the openings may have a significant effect on where the effective center would be located on the pitorifice.

As the pitorifices are inserted deeper into the pipe, more flow is obstructed and the influence of the downstream pitorifice becomes greater. This can be seen clearly with the decreasing values in Table 7.3 which gives the ratio of K_{up} to K_{po} (for $x_i=0$ the mean values are used).

Table 7.3: K_{up}/K_{po}s for Pitorifices at Different Insertion Depths

Nom. Main	Nom. PO	Type PO	K_{up}/K_{po} -3.0	-1.5	0	1.5	3.0
6"	3/4"	BC&T	0.62	0.62	0.59	0.55	0.53
6"	3/4"	IB&C	0.57	0.59	0.59	0.58	0.53
6"	3/4"	IBC&T	0.60	0.60	0.59	0.54	0.53
6"	3/4"	C	0.43	0.39	0.36	0.33	0.29
6"	3/4"	C&B	0.61	0.60	0.58	0.56	0.52
6"	**3/4"**	**L**	**0.71**	**0.69**	**0.67**	**0.64**	**0.60**
6"	3/4"	LF	0.67	0.66	0.64	0.60	0.59
6"	3/4"	45L		0.57	0.56	0.55	0.53
6"	1"	BC&T	0.56	0.54	0.52	0.48	0.49
6"	1"	IB&C		0.55	0.53	0.51	0.49
6"	1"	IBC&T	0.49	0.53	0.51	0.49	0.46
6"	1"	45L			0.52	0.50	0.49
4"	3/4"	BC&T			0.49		
4"	3/4"	SQ			0.52		0.51
4"	3/4"	45L			0.48		0.45
4"	1"	BC&T			0.41		
4"	1"	IB&C			0.41		
4"	1"	SQ			0.51		

The ¾-inch pitorifices in the 6-inch diameter pipe offer the best basis of comparison because the influence of the pitorifice is less than with the 1-inch pitorifices or 4-inch diameter pipe. All the pitorifices used also have different projected areas within the pipes for a given insertion. Increasing the projected area can be expected to increase the value measured for K_{po} by increasing the velocity past the downstream pitorifice. This projected area causes a flow constriction which accelerates flow and causes the static head to drop and increases the influence of the pressure drag effect. The flow constriction also increases the frictional drag losses. These effects were addressed in Equation 4.15 which is rewritten here:

$$K_{po} = K_{up} - C_p + C_f^* \tag{7.5}$$

where terms have been previously defined. Assuming C_f^* is negligible, C_p can be found from K_{po} and K_{up}. Table 7.4 gives K_vs which give some qualitative idea of the contribution that might be due to blockage effects and Table 7.5 gives calculated C_ps. The L shape is one of the best shapes for the upstream pitorifice as can be seen by comparing values for K_{up} in Table 7.2, but it is the worst shape in the downstream position as can be seen by comparing values for C_p in Table 7.5. The result is a relatively poor overall performance as seen in Table 7.1. The IB&C shape is one of the most effective shapes for overall performance and is a little more effective than the IBC&T shape for the 1-inch size. It has significantly better upstream performance which more than offsets poorer downstream performance. Only the 1-inch sizes can be compared this way because the ODs for the ¾-inch sizes are significantly different, as shown in Table 6.1.

Table 7.4: K_vs for Pitorifices at Different Insertion Depths

Nom. Main	Nom. PO	Type PO	K_v -3.0	-1.5	0	1.5	3.0
6"	3/4"	BC&T	0.10	0.14	0.19	0.23	0.28
6"	3/4"	IB&C	0.11	0.16	0.20	0.25	0.30
6"	3/4"	IBC&T	0.09	0.13	0.17	0.22	0.27
6"	3/4"	C	0.11	0.16	0.20	0.25	0.30
6"	3/4"	C&B	0.07	0.11	0.15	0.20	0.25
6"	**3/4"**	**L**	**0.13**	**0.17**	**0.22**	**0.27**	**0.32**
6"	3/4"	LF	0.13	0.18	0.22	0.27	0.33
6"	3/4"	45L	0.11	0.16	0.21	0.26	0.31
6"	1"	BC&T	0.13	0.19	0.25	0.32	0.39
6"	1"	IB&C	0.13	0.18	0.24	0.31	0.38
6"	1"	IBC&T	0.12	0.18	0.24	0.30	0.38
6"	1"	45L			0.28	0.35	0.42
4"	3/4"	BC&T			0.27		
4"	3/4"	SQ			0.35		0.62
4"	3/4"	45L			0.32		0.59
4"	1"	BC&T			0.38		
4"	1"	IB&C			0.36		
4"	1"	SQ			0.48		

Table 7.5: C_ps for Pitorifices at Different Insertion Depths

Nom. Main	Nom. PO	Type PO	C_p -3.0	-1.5	0	1.5	3.0
6"	3/4"	BC&T	-0.68	-0.75	-0.89	-0.90	-0.90
6"	3/4"	IB&C	-0.80	-0.85	-0.92	-0.96	-0.94
6"	3/4"	IBC&T	-0.74	-0.78	-0.89	-0.90	-0.90
6"	3/4"	C	-0.75	-0.95	-1.02	-1.04	-1.03
6"	3/4"	C&B	-0.65	-0.76	-0.87	-0.87	-0.87
6"	**3/4"**	**L**	**-0.49**	**-0.58**	**-0.67**	**-0.68**	**-0.67**
6"	3/4"	LF	-0.57	-0.65	-0.78	-0.78	-0.78
6"	3/4"	45L		-0.88	-0.94	-0.95	-0.95
6"	1"	BC&T	-0.81	-0.98	-1.15	-1.16	-1.16
6"	1"	IB&C		-0.96	-1.15	-1.16	-1.16
6"	1"	IBC&T	-0.88	-1.00	-1.18	-1.19	-1.19
6"	1"	45L			-1.14		-1.14
4"	3/4"	BC&T			-1.21		-1.21
4"	3/4"	SQ			-1.08	-1.08	-1.08
4"	3/4"	45L			-1.26	-1.27	-1.27
4"	1"	BC&T			-1.52	-1.51	-1.52
4"	1"	IB&C			-1.55	-1.58	-1.57
4"	1"	SQ			-1.02	-1.03	-1.03

7.2.6 Head Loss in Pitorifice Pairs

To determine the head loss in a service line due to losses in the pitorifices, service line flow data were also collected by data logging flow rates in the service loop. Testing was done at V_m=0.9, 0.67, and 0.3 m/s and $x_i = -3$, 0, and 3 cm. Flow was throttled in the service loop to span the laminar flow regime. The head loss due to the pitorifices was determined by subtracting the sum of the observed head and the calculated loss in the length of tube between piezometers from the calculated shut-off head for the V_m and x_i used in the test run. Resulting head losses were then used to calculate a fitting loss for the pitorifices alone, $K_{ftg,po}$ or K_f, using Equation 4.23. The graphs for K_fs are given in Appendix B. The K_fs were found to be higher at the lower Reynolds numbers, similar to the K_{ftgs} shown in Figure 7.2. An equivalent length for the pitorifices alone, $L_{eq,po}$, was also found by rearranging Equation 4.20:

$$L_{eq,po} = \frac{g \rho \pi D_s^4 H_{ftg,po}}{128 \nu \dot{m}_s} \qquad (7.6)$$

where $H_{ftg,po}$ is the loss attributable to the pitorifices alone. Equivalent length appeared to give the best correlations, although the data scatter due to the extremely low head differentials made correlations difficult to determine. The equivalent length appears to be fairly independent of both the service line Reynolds number and the insertion depth of the pitorifice, but may be dependent on the Reynolds number in the main. Figure 7.13 shows L_{eq}s calculated for 1-inch pitorifices. At the lower flow velocities in the main, data scatter is less because pressure fluctuations are less, and L_{eq} appears to be lower in value. The means are 3, 8, 11, and 28 m for Re_m=48,000, 78,000, 106,000, and 143,000, respectively.

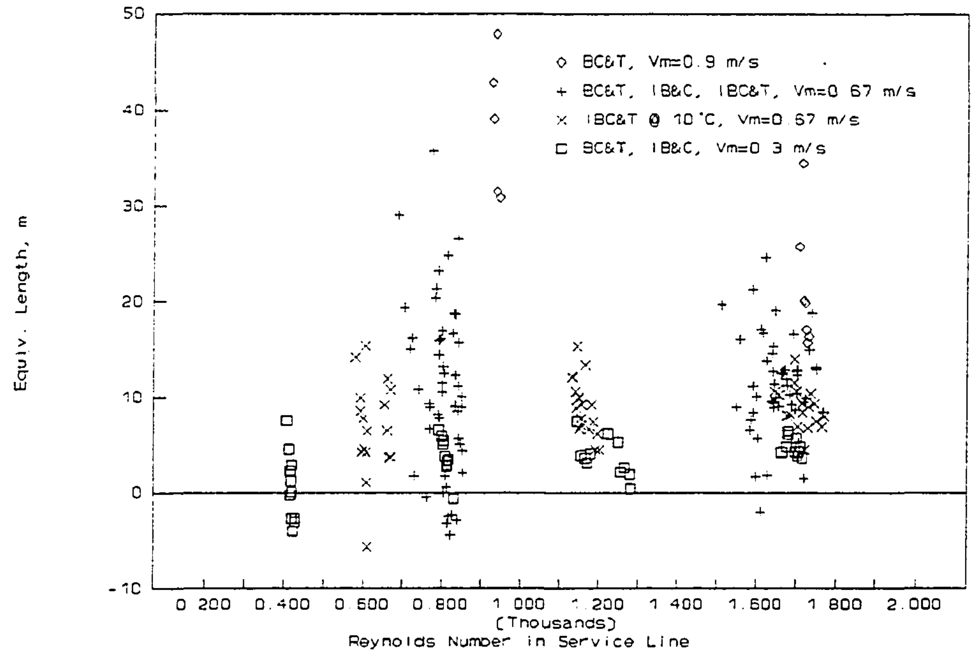


Figure 7.13: Equivalent Length for 1-inch Pitorifices in 6-inch Pipe

Figure 7.14 shows L_{eq}s calculated for ¾-inch pitorifices. The means for the various bent shapes are 1, 7, and 11 m and for the L and LF shapes they are 0.4, 5, and 5 m for Re_m=48,000, 106,000, and 143,000, respectively.

7.2.7 Loss in Pitorifice Performance

It was convenient to assume a constant head from a pitorifice pair for a given flow in the main and assume performance and head losses in the pitorifices to be part of the service line losses. However, calculating the effective K_{po} when water is flowing in a service is helpful in understanding the magnitude of these losses. The ratio of the effective K_{po} with flow in the service to the K_{po} for the case with no flow in the service can be considered a K_{po} performance loss coefficient. Figure 7.15 shows this loss coefficient as a function of Reynolds number in the service line. In a sense, this is a dimensionless performance curve analogous to a dimensionless pump performance curve and it allows a design engineer a method of derating the assumed K_{po} at shut-off for a given flow. This would be more cumbersome than adding an equivalent length to the service line length to account for performance loss.

The data scatter seen in Figure 7.15 also serves to illustrate why some of the calculated equivalent lengths in Figures 7.13 and 7.14 are negative.

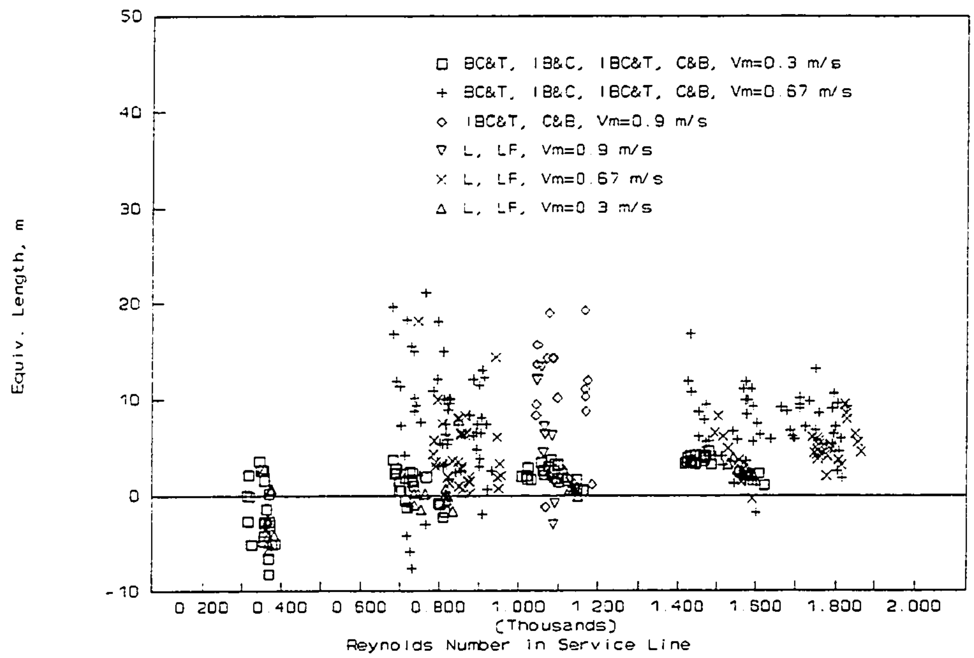


Figure 7.14: Equivalent Lengths for ¾-inch Pitorifices in 6-inch Pipe

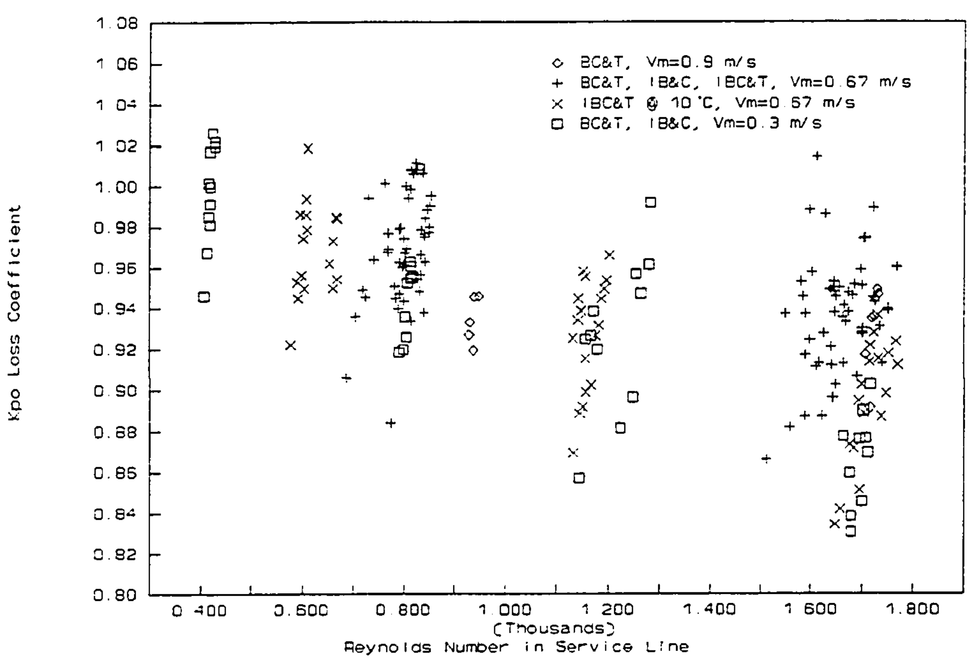


Figure 7.15: Performance Loss Coefficients for 1-inch Pitorifices in 6-inch Pipe

7.3 REDUCED SCALE PITORIFICE PERFORMANCE TESTING

The objective of reduced scale testing was to provide more detailed information on pitorifice performance by allowing a wide range of relative sizes with the same shape to be evaluated. The 45L pitorifice was chosen because it was representative of the shapes historically used and because it was also easily fabricated in different sizes. All tests were done at V_m=0.7 m/s and K_{po} and C_f were calculated using mean values for the measured differential heads instead of a linear regression as was done in the full scale tests. Nominal ⅜, ½, ¾, and 1-inch pitorifices having stem ODs of 0.500, 0.625, 0.875, and 1.125 inches, respectively, were used in 4 and 6-inch diameter main test sections. The two larger sizes for pitorifices were not used in the 4-inch diameter main because of their limited travel. Figure 7.16 shows the results with the insertion depths normalized by reporting them as a fractional distance from the center to the ID of the main. A line showing the head from an ideal pitot tube is also shown. Note that the values for K_{up} are in approximate agreement with this line if allowance is made for the position of the effective opening on the pitorifice relative to the tip.

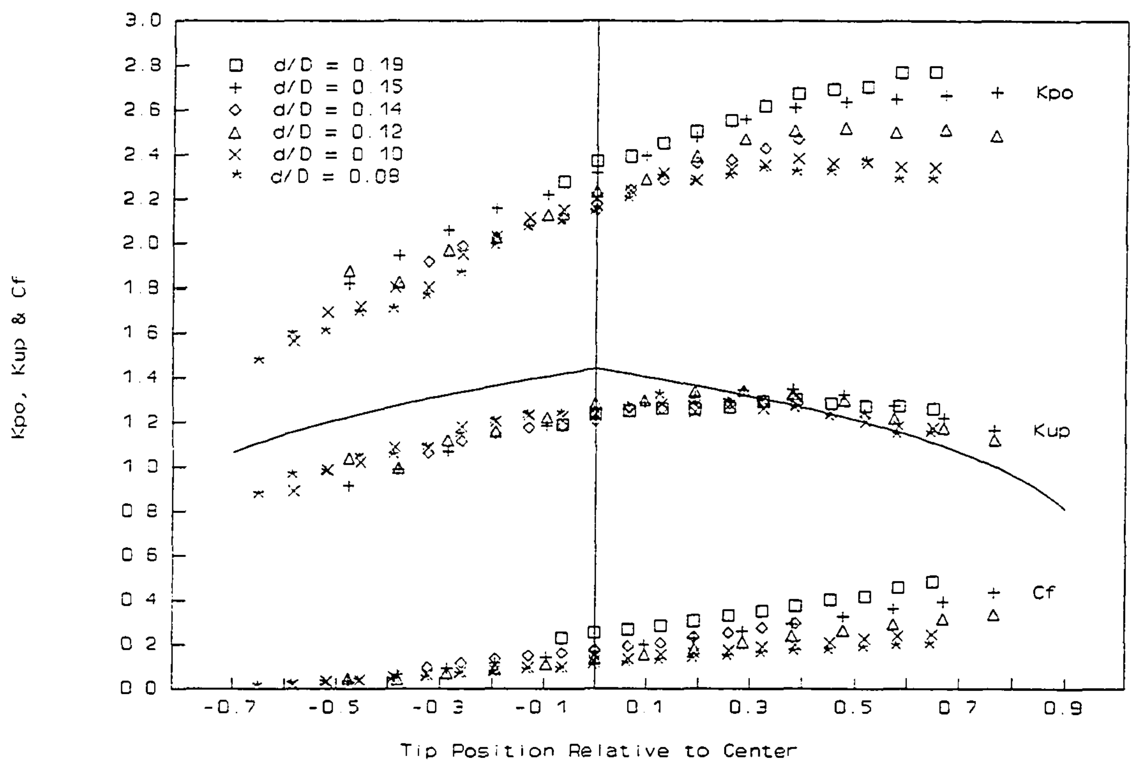


Figure 7.16: Performance of 45L Pitorifices

7.4 CONCLUSIONS

The 45L pitorifice is fairly representative of the shapes historically used. The C, L, LF, and SQ shapes are not as effective as the other shapes. Insertion depth has the greatest impact on performance. Head loss in the main due to pitorifices is equivalent to only a couple meters of main. Head loss in the service loop due to pitorifice friction and performance losses is equivalent to about ten meters of service line. There is a small decrease in performance as the pitorifices are moved closer together.

CHAPTER 8: TESTING OF SMALL CIRCULATION PUMPS

The smallest of the pumps commonly used to circulate water in a service loop require about 85 watts of electric power to deliver less than 10 watts of hydraulic power for a wire-to-water efficiency of about 12 percent. Where electric rates are $0.42/kWh this results in a cost of $24 per month for the individual or, for a community with 100 pumps operated year round, about $30,000 per year. In most applications less than two watts of hydraulic power is required for freeze protection, so more efficient and smaller pumps will allow considerable savings.

Pump requirements, pumps manufactured by ten different companies, speed control, and no-flow alarms are discussed in the following sections. Pump performance curves supplied by the manufacturers for 15 pumps are shown with their nominal wattages along with representative service line system curves. Test results for three pumps are also given.

Prevailing electric rates should be considered as well as the first cost of a pump. At $0.42/kWh, a 40 watt savings equates to over $1,400 in ten years. The potential savings over the expected life span for a pump are much larger than the installed cost of the pump. These issues are dealt with in greater detail in Chapter 10.

8.1 PUMP REQUIREMENTS

Maintaining a minimum flow rate in a service line is the most important requirement, but minimizing energy costs and maximizing service life are also important.

The required flow rate is a function of the heat loss rate. For a simple estimate of worst case required flow, assume the service lines are contained in an above-ground insulated conduit and that the drop in temperature of the water is small compared to the temperature differential across the insulation. Then the following approximation can be made:

$$\dot{E}_h = \frac{\Delta T_I L_C 2\pi k_I}{\ln[(D+2t)/D]} = \Delta T_W Q \rho C_p \tag{8.1}$$

where $\dot{E}_h$ is the rate of heat energy transfer to the environment, ΔT_I is the temperature differential across the insulation, L_C is the conduit length, k_I is the thermal conductivity of the conduit insulation, D is the diameter of the conduit, t is the insulation thickness, ΔT_W is the temperature drop of the water in the line, Q is the volumetric flow rate in the service line, and ρ and C_p are the density and heat capacity of water. Equation 8.1 shows that Q is directly related to L_C, k_I, and ΔT_I, and inversely related to ΔT_W. The relationship between Q and D and t is more complex; Figure 8.1 shows the required thickness of polyurethane insulation with k_I=0.03 W/m·C° [0.017 Btu/hr·ft·F°] for an above ground conduit as a function of Q assuming L_C=50 m, D=100 mm, ΔT_W=2 C°, and ΔT_I=50 C°. A flow of about 0.1 L/s for an insulation thickness of t=40 mm is required for these rather extreme thermal conditions. Figure 8.1 and Equation 8.1 can be used to quickly estimate the required flow for other conditions; for example, if t=60 mm, Q must be 0.07 L/s at the initially assumed conditions or 0.02 L/s at ΔT_I=25 C° and L_C=30 m, other things being equal. The volumetric flow will usually need to be in the range of 0.01 to 0.20 L/s [0.2 to 3.2 gpm] and will often be much less than 0.1 L/s [1.5 gpm], especially for buried conduits.

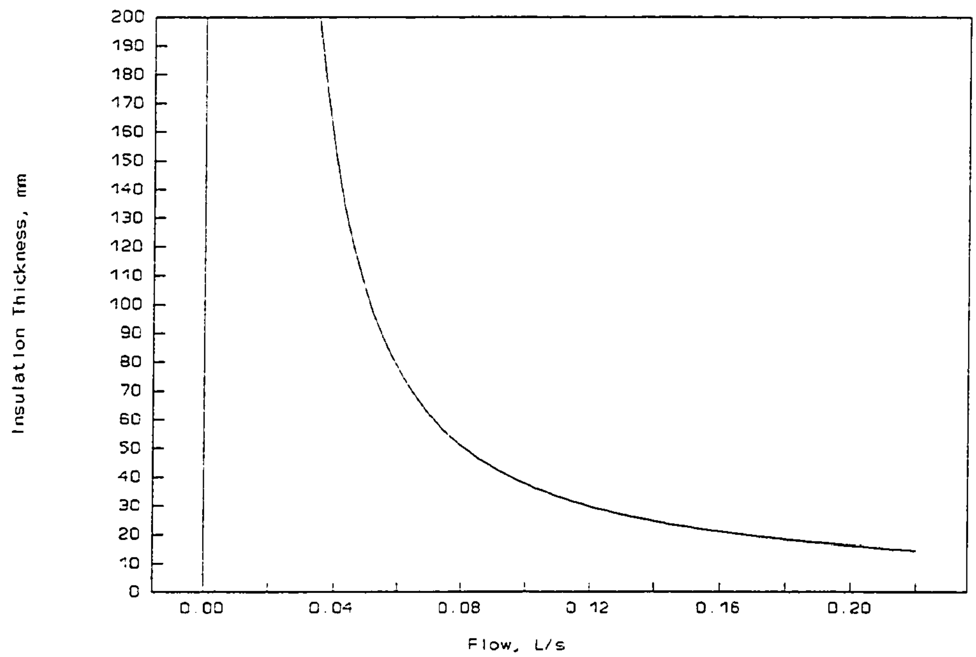


Figure 8.1: Required Insulation Thickness as a Function of Flow Rate

The head requirements are a function of the total service line length which is twice the distance from the main to the house (L_S=2L_C), its hydraulic roughness (ϵ), its internal diameter (d), the total fitting loss factor for the line (ΣK_{ftgs}), and the kinematic viscosity of the water (ν). Figure 8.2 shows system curves for service lines with $\epsilon=1.5\times10^{-6}$, d=20 mm, ΣK_{ftgs}=5, and L_S=100 m and L_S=50 m and water at 5°C with $\nu=1.5\times10^{-6}$ m²/s . Note the transition from a linear relationship in the laminar range to an exponential one in the turbulent range. Head requirements will be significantly less in larger diameter service lines. Figure 8.2 also shows curves of constant hydraulic power.

The longer line will require twice the flow of the shorter line assuming the same heat loss rates per unit length and the same allowable drop in water temperature. For the above assumptions, the head

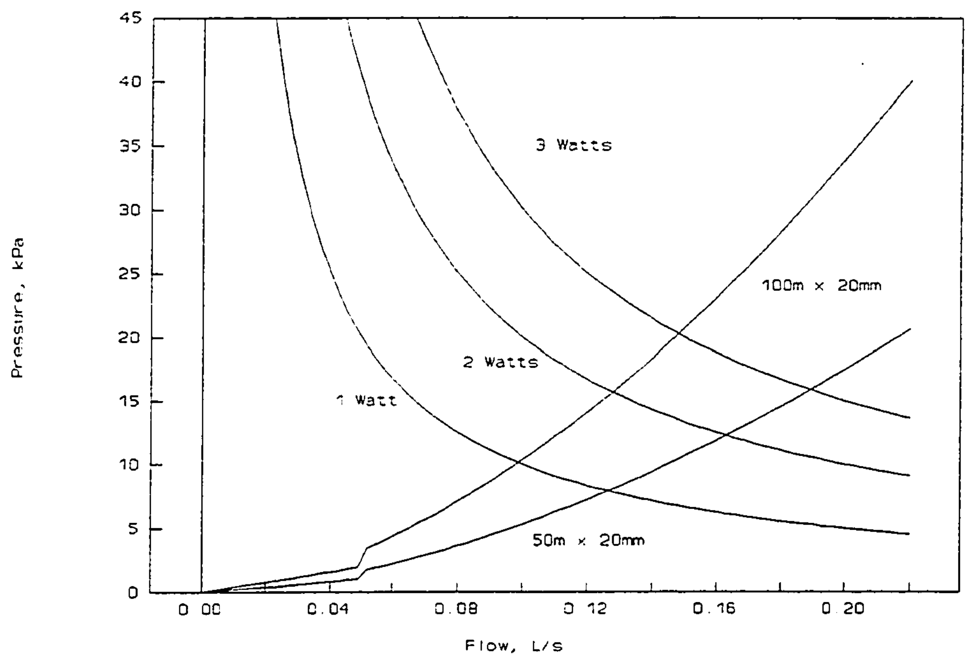


Figure 8.2: Hydraulic Power and Service Line System Curves

loss in the longer line will be 10 kPa at 0.1 L/s whereas that for the shorter line will only be about 2 kPa at 0.05 L/s. Based on this it can be assumed that the required pump operating head will typically be between 1 and 20 kPa [0.3 and 6.6 ft WC].

The energy consumption of a pump is a function of the hydraulic power requirement and the pump and motor efficiencies. Small pumps and motors will generally have efficiencies of less than 40 and 60 percent, respectively, for a combined wire-to-water efficiency of less than 24 percent, so power requirements will generally be greater than one watt and as much as 20 watts or more at the upper end. Wire-to-water efficiencies give some insight into pump performance, but while a good wire-to-water efficiency is desirable, it is also important to obtain a good match between the pump and the minimum flow requirements. A poorly selected efficient pump which is supplying significantly more flow than needed can use much more energy than an inefficient pump supplying only the required flow.

In addition to meeting the flow requirements, there are construction considerations. The pump should be able to withstand system pressures (often in excess of 830 kPa [120 psi]), the inlet and outlet should be in line and compatible with service line sizes (nominal ¾-inch diameter and larger), and all parts in contact with water should be non-toxic and made of corrosion resistant materials such as stainless steel, bronze, or a plastic. Quiet operation and no lubrication or other maintenance requirements are highly desirable. It may be advantageous to use heat from the motor to warm the water being pumped to reduce system heating requirements. The pump should permit demand flows which may greatly exceed the open discharge flow to pass without damage; ideally these flows should be allowed to go either direction. If the pump is to be used in conjunction with pitorifices, it is desirable that it allow flow through the housing and impeller with minimum head loss when not operating. The pump will be exposed to water which will often be at temperatures below ambient. Condensation on the pump body may occur when the pump is not running and the motor is not generating heat. Therefore the pump and motor should be coated and sealed as needed to prevent corrosion damage. This is particularly important if the pump is to be run only seasonally and will be inoperative for long periods of time. The pump should have a high starting torque if it is operated seasonally where deposits may form, and the ability to run dry may be an advantage in some situations. Finally, the cost of owning and operating the pump should be considered.

Pumps which approach these criteria are generally intended for hydronic heating, solar heat collection, potable water heat exchange, or potable hot water circulation. They have either a magnetic coupling between the pump impeller and the motor or a motor rotor which is connected directly to the pump impeller. Both the magnetic coupling and the wet rotor designs minimize problems with seals and bearings. The wet rotor pumps transfer much more motor heat to the water being pumped and, unlike many magnetically coupled pumps, they do not require lubrication.

Hydronic heating systems are usually closed systems which recirculate the same water. The water in a closed system characteristically has a low dissolved oxygen content and may also have chemical inhibitors; this allows the use of cast iron in pump construction. Systems intended for heating water meant for consumption are open systems because the water is constantly renewed. The pumps in these systems need to be made of corrosion resistant materials.

Water is usually circulated at relatively high velocities in a closed loop when heat exchange is involved to minimize capital costs for the pipe and heat exchanger, whereas relatively low circulation rates are acceptable for service line freeze protection. The system which most closely resembles arctic service line circulation in operating requirements is potable hot water circulation. In potable hot water circulation, water in the hot water distribution lines in a building is kept hot at the point of use by providing a return line and a pump. The higher heat loss rate, the additional electric cost, and additional capital costs for the extra pipe, fittings, and pump will often be offset by water use savings. These systems have been used primarily for consumer comfort and convenience, particularly where very long runs are used such as in hotels. In recent years the use of potable hot water circulation for water conservation has become more important particularly in places where water supplies are limited such as in California and where water is hauled and not piped to homes. An example of the later application is described by Hostland et al. (1994).

Most of the smaller pumps use 115 VAC motors with power use that is relatively constant over the entire operating range. Gerlek (1982a and 1982b) made power measurements on pumps from three different manufacturers (Grundfos, Taco, and Armstrong) and found power variation over the pump operating ranges to be less than ten percent in all cases. Small shaded-pole (SP) AC motors are only about 20 percent efficient and permanent-split capacitor (PSC) AC motors are about 40 percent efficient. Small DC motors can be 60 to 80 percent or more efficient but this is offset by AC to DC conversion efficiencies of 50 to 90 percent for power supplies. Most DC motors available at this time use brushes which would require annual or biannual adjustment and replacement when operated continuously, so they are not suitable for use in homes unless they are maintained by the utility. Brushless DC motors are becoming more common because of their high efficiency and versatility, but they cost more than AC motors and only two of the pump manufacturers surveyed have models that are available with a brushless DC motor or driver.

8.2 PUMP MANUFACTURERS

Ten different pump manufacturers were investigated. The first four presented in the following subsections represent the bulk of the small pumps used for service line circulation. Pumps from the next four manufacturers have never been used for that purpose prior to this study to the best of the author's knowledge. The last two manufacturers supply pumps which are representative of small pumps not suitable for service line circulation.

Information on pumps using 50 watts or less is summarized in Table 8.1. The heads and flows listed were found using the manufacturers's pump curves and a system curve calculated for 100 m [328 ft] of 20 mm [0.79 inch] ID smooth pipe. More detailed information on these and other pumps is given in the following subsections. Refer to Appendix G for selected pump curves in USCS units of head in feet of WC and flow in gpm and Refer to Appendix D for general information on pumps and motors.

8.2.1 Grundfos

Grundfos (Clovis, California) manufactures wet rotor pumps designed for open system water applications which can be operated up to system pressures of 980 kPa [142 psi]. Their model UP15-18 series, available with either bronze or stainless steel pump volute (indicated with a B or S at the end of the model designation), was their lowest flow open system pump until 1993 when the model UM15-10 series, available only with bronze volute, was introduced. Their UP and UM models are 2 pole and 4 pole, nominal 3,600 and 1,800 RPM respectively. Figure 8.3 shows the performance curves for the UP15-18S and UP15-42S models with the UM15-10B along with the assumed system curves. All three pumps are available with ¾-inch fittings, and the UP15-42 is also available with 1-inch fittings. The power head assemblies and volutes for pumps manufactured around the same time are interchangeable but not sold separately. The only difference between the UP15-18 and the UP15-42 appears to be the impeller; the UP15-18 has an open impeller and the UP15-42 (like the UM15-10) has a closed impeller. A bronze volute on either the UP15-18 or the UP15-42 significantly improves performance.

Table 8.1: Pumps Using 50 Watts or Less

Pump Model	Type	Watts	KPa	L/s	Eff.	Notes
Grundfos UM15-10B	WR	38	12.5	0.110	4	1
Hartell WR	WR	50	15.5	0.125	4	2
Hartell MD	M	30	8.0	0.085	2	3
Hartell HEL	M	8	12.5	0.110	17	4
Laing 303	WR	33	17.0	0.135	7	5
Little Giant CMD-100-B	M	34	8.5	0.090	2	3
March 809	M	30	8.5	0.090	3	3
March 809/Ivan Labs	WR	4	4.0	0.060	6	6

1. WR is wet rotor, M is magnetic drive, and efficiency is wire-to water in percent.
2. Unsealed windings may be susceptible to condensation.
3. Should be plumbed upstream of user tap unless check valve used to prevent backflow.
4. Brushless DC motor with 12 V operation. Power supply losses are not included.
5. Should be plumbed downstream of user tap with check valve to prevent backflow.
6. Brushless DC driver with 17 V operation. Power supply losses are not included.

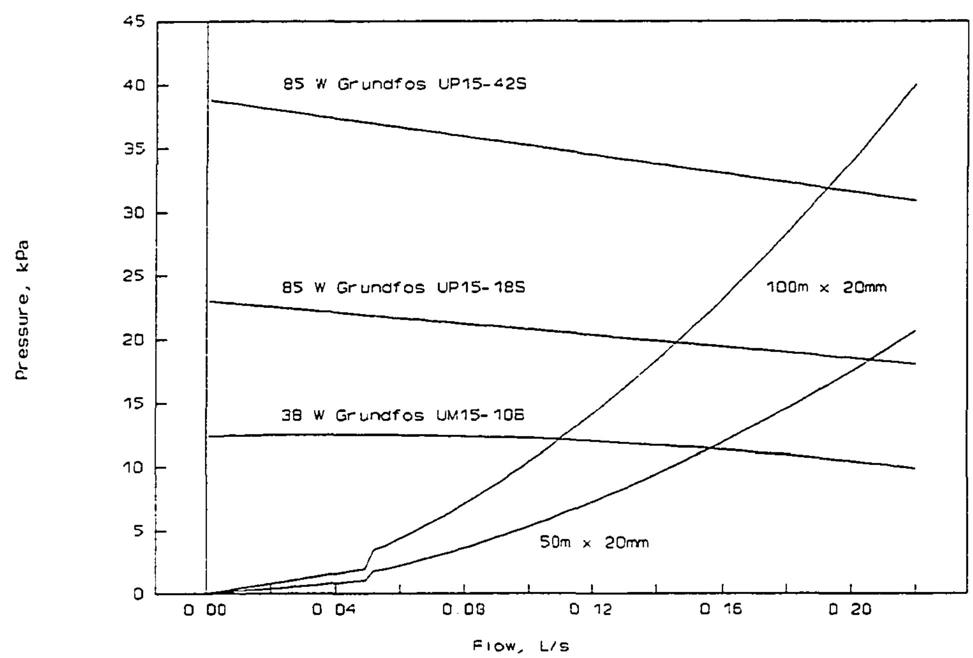


Figure 8.3: Grundfos Pump Performance Curves

Grundfos offers the UP models for closed systems as well, with the UP15-42 available as a 3-speed model. This model has been used in potable water systems although it has a cast iron volute which is subject to corrosion. Speed control is done by switching the connection points between the run and start windings to allow more slip. At the lowest speed the power requirement is 45 watts and the performance is better than the UM15-10, but the start-up torque is less. Grundfos advises starting the pump at the highest speed after it has been out of operation for a season. Presumably, the 3-speed power head could be installed on a bronze or stainless steel volute or the 3-speed switch assembly could be installed on a UP15-42 (or UP15-18 for that matter) but Grundfos does not support this.

Grundfos used to impregnate the windings in their pumps with a sealant but they have not been doing this since the early 1980s (Garroute, 1994) and so they do not guarantee the pump when it is used to pump water colder than 50°F [10°C]. However, the windings are contained within a housing which is sealed by the connection plug so the risk of water vapor condensing on the coils is not great.

Figure 8.4 shows wire-to-water efficiencies calculated for four different model Grundfos pumps using published performance curves and assuming constant wattage input to the pumps. The UP15-42B has a peak wire-to-water efficiency of 20 percent. Grundfos claims a pump efficiency of 35 to 40 percent for these small pumps (Garroute, 1992) so the motor efficiency should be about 50 percent. This is a reasonable value because the pumps use permanent-split capacitor motors.

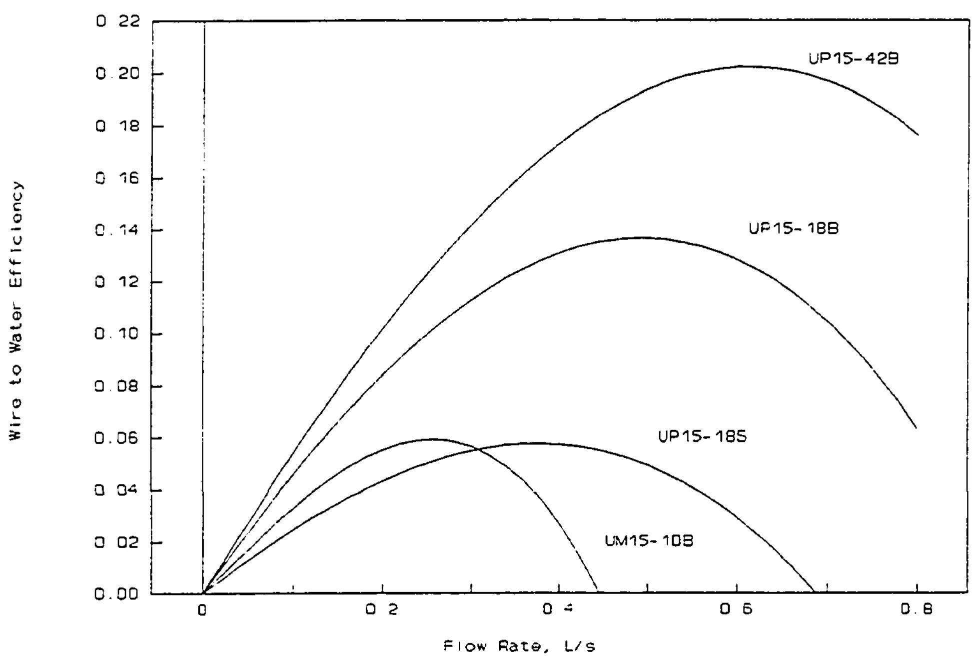


Figure 8.4: Wire-to-Water Efficiencies for Selected Grundfos Pumps

The only difference between the model UP15-42B and the UM15-10B is speed. Theoretically, for an ideal pump, there should be no loss in efficiency with decreased speed, but this is clearly not the case because the UM15-10B has a wire-to-water efficiency of six percent, perhaps due primarily to a loss in motor efficiency. The lower efficiencies of the other two pumps may be attributed to an open impeller in the case of the UP15-18B and both an open impeller and a stainless steel volute with greater clearances and a less efficient shape in the case of the UP15-18S.

It is important to understand the factors that contribute to high efficiency, but pump selection should be based on the total electric power requirement and not efficiency alone. If a pump delivers more flow than required, the head will also be higher than required and the power requirement can be much higher than a more appropriately sized but less efficient pump.

8.2.2 Armstrong

Armstrong Pumps (North Tonawanda, New York) makes small two-speed wet rotor pumps which are very similar to the Grundfos pumps in construction. The performance curve for their smallest size circulator, model SS-30B, supplied with flanges ranging from ¾ to 1½-inch and running at the low speed setting (1,900 RPM) is shown in Figure 8.5. The high speed setting (2,900 RPM) uses 85 watts and the low speed setting uses about 70 watts for a wire-to-water efficiency of nine percent.

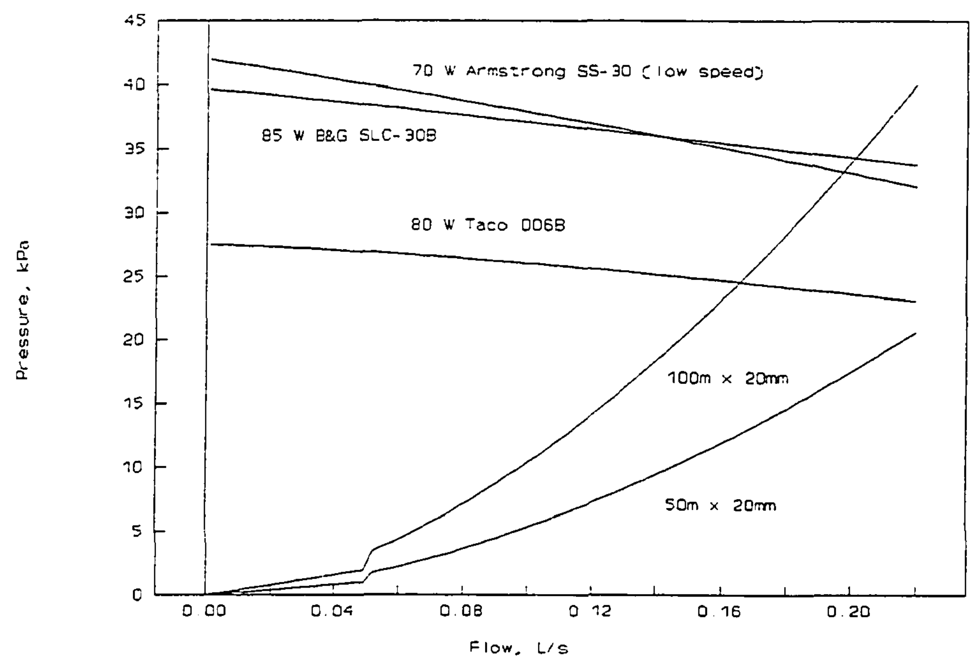


Figure 8.5: Armstrong, B&G, and Taco Pump Performance Curves

8.2.3 Bell & Gossett

ITT Bell & Gossett (Morton Grove, Illinois) manufactures several circulating pumps, the smallest being the wet rotor, bronze body Red Fox Model SLC-30B which uses 85 watts and has ¾ to 1½-inch fittings. The pump has a maximum operating pressure of 860 kPa [125 psi] and is similar to the Grundfos pumps. See Figure 8.5 for the pump performance curve.

8.2.4 Taco

Taco Pumps (Cranston, Rhode Island) manufactures several circulating pumps, the smallest being their Model 006B with ¾-inch connections. It is similar in design to the Grundfos pumps except that the wet rotor assembly can be replaced as a unit instead of the whole pump assembly. The pump performance is shown in Figure 8.5.

8.2.5 Hartell

Hartell (Ivyland, Pennsylvania) manufactures AC wet rotor and magnetic drive pumps and DC brush-style and brushless magnetic drive pumps. The AC wet rotor pump windings are exposed to the atmosphere so they may be susceptible to damage from condensation when there is cold water in the lines and the pump is not operating. Figure 8.6 shows the performance of their AC pumps and the performance ranges of two brushless DC pumps. The brushless DC motors are available in high and low speed versions and can be powered by different DC voltage level supplies to match performance to requirements.

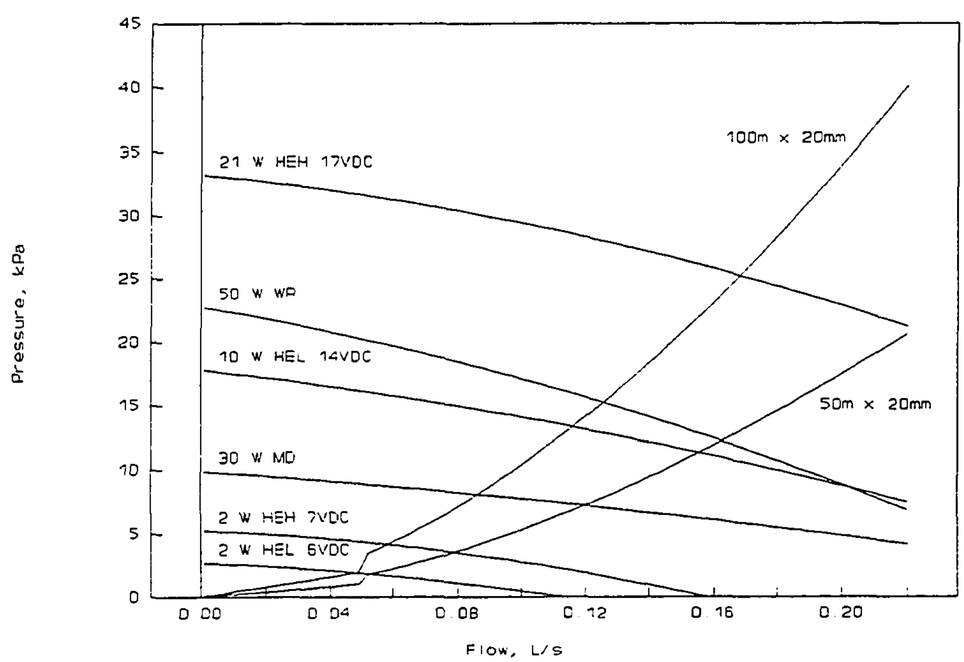


Figure 8.6: Hartell Pump Performance Curves

8.2.6 Laing

Laing Thermotech Inc. (Chula Vista, California) makes small pumps intended for use in domestic hot water circulation systems. Their smallest AC model, the 303B, uses a wet rotor and a special semi-spherically shaped synchronous motor armature. Their DC model, the 12 VDC MC-201DC-B, uses a cup shaped magnetic coupling on the shaft of a standard brush type DC motor to drive a magnetic impeller sealed in the pump housing. Both pumps have a maximum working pressure of 1,000 kPa [150 psi] and come with ½-inch fittings. Figure 8.7 shows the pump performance curves. The 12 VDC pump wire-to-water efficiency is about twice that of the AC pump at the operating point, probably due primarily to the higher efficiency of small DC motors. Unfortunately, Laing rates the DC pump life at only 10,000 hours versus 100,000 hours for the AC pump. Laing also recommends plumbing their pump downstream of the supply tee with a check valve to prevent back-flow.

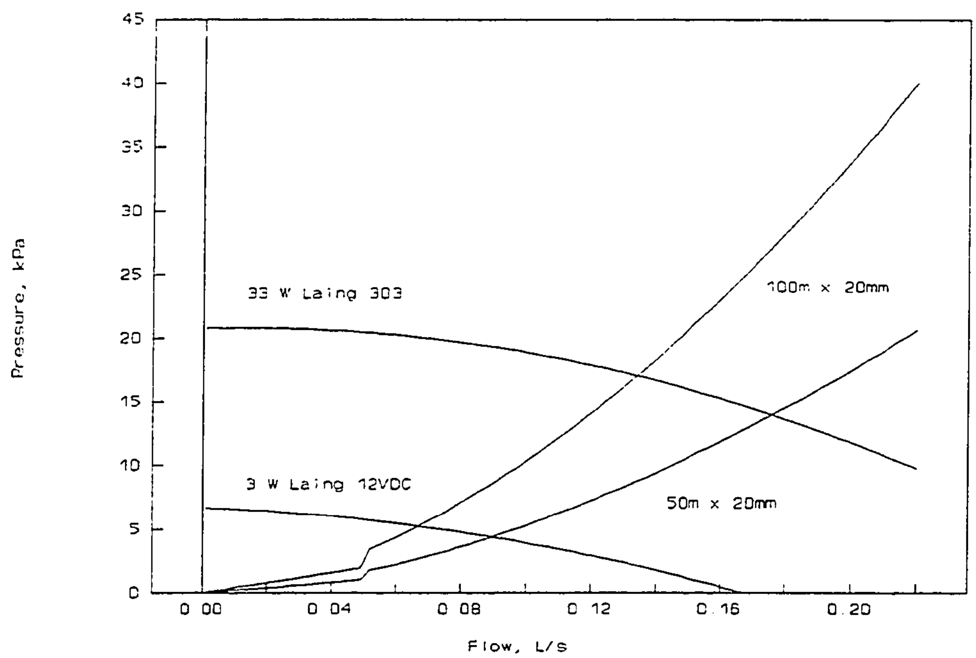


Figure 8.7: Laing Pump Performance Curves

8.2.7 Little Giant

Little Giant Co. (Oklahoma City, Oklahoma) manufactures a small magnetically coupled pump, the Model CMD-100-B, which is intended for hot water circulation in solar, heat recovery or hydronic applications. The pump is driven by a shaded-pole AC motor built to their specifications. It is designed for a maximum system pressure of 1,000 kPa (150 psi) and has ½-inch inlet and outlets. The motor uses only 34 watts and runs at 1,550 RPM. Unfortunately, the manufacturer does not supply this model pump with a permanent-split capacitor motor. Such a motor would use about one-half the energy based on the other pump models which are supplied with either an shaded-pole or a permanent-split capacitor motor. Figure 8.8 shows the performance curve for this pump.

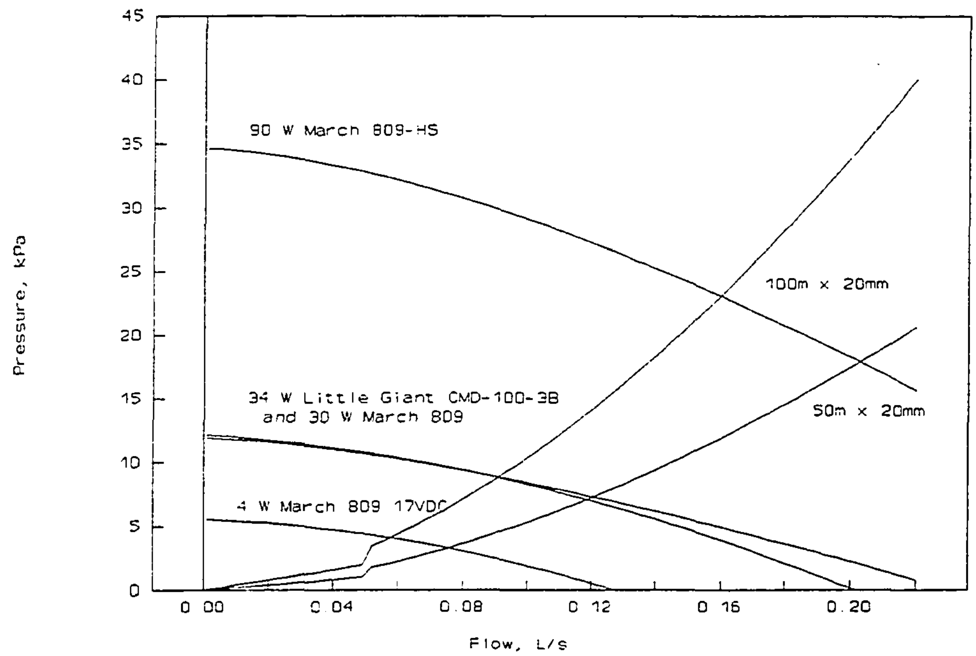


Figure 8.8: Little Giant and March Pump Performance Curves

8.2.8 March

March Manufacturing Inc. (Glenview, Illinois) manufactures several small magnetically coupled pumps with 115 and 24 Volt AC shaded-pole and 12 or 24 Volt brush-style DC motors which provide a variety of speeds and performances. Ivan Labs Inc. (Jupiter, Florida) makes a brushless DC driver for the March 809 pump. The performance curves of the March Model 809 driven at both 1,700 and 3,400 RPM (Model 809HS for "high speed") with AC motors and fitted with the Ivan Labs driver are shown in Figure 8.8. Unlike the Hartell pumps which are supplied with brushless DC motors which in turn drive a magnetic coupling, the Ivan Labs driver drives the impeller directly, analogous to a wet rotor AC motor/pump combination. They also offer a driver which produces about 15 percent higher head.

8.2.9 Micropump

Micropump Corp. (Vancouver, Washington) manufactures a number of small gear displacement pumps and one centrifugal pump. This company is representative of a number of firms which supply pumps for the chemicals industry. The gear displacement pumps have very close tolerances and are unsuitable for a fluid which may contain particles. Corrosion particles are a possibility with domestic water supplies. Their Model 101-605 centrifugal pump is magnetically coupled to a 3,000 RPM shaded-pole motor. The pump is suitable for system pressures of 1,400 kPa [200 psig] but reportedly uses 1.9 A at 115 V (or about 200 watts assuming a 0.9 power factor). In addition to its high power requirement the pump has a ⅜-inch female pipe thread (FPT) inlet and ¼-inch FPT outlet which are considerably smaller than the line sizes being served, and it costs substantially more than pumps intended for hydronic systems.

8.2.10 Flotec

Flotec (Leavan, Wisconsin) manufactures a small flexible vane positive displacement pump powered by a universal motor. According to the manufacturer, the seals on these pumps will leak at line pressures above 240 kPa [35 psi]. This company is representative of a number of firms which supply low cost transfer pumps. These pumps are not intended for continuous duty.

8.3 POTENTIAL PUMP MODIFICATIONS AND OPTIONS

All of the pumps currently on the market supply much more than the required flow for most service line freeze protection situations and most of the motors used are very inefficient. The most obvious modifications are reducing output and increasing motor efficiency. Output can be reduced by reducing impeller diameter or speed. Reducing impeller diameter will reduce pump efficiency but will still result in a net reduction in power requirements, allowing a smaller motor to be used. Speed can be reduced by substituting a motor with more poles or by using speed control. Speed control can be built into a motor by tapping the windings so that selective switching can be done to weaken the magnetic field which in turn allows more slip during operation. Speed control can also be done by add-on devices which switch AC or DC power on and off, limit the voltage of AC or DC supply, or reduce the frequency of AC power.

The effect of speed reduction on a pumps performance can be estimated by the similarity rules which predict changes in flow, head, and power to be proportional to the first, second, and third powers of the change in speed respectively. Referring to Figure 8.8 which shows the performance of the March Model 809 pump at 1,700 and 3,400 RPM it can be seen that the similarity rules are far from strictly applicable. In this case reducing speed by half cuts head by one-third, not one-fourth.

Motors for small pumps may be available with different enclosures, ball bearings, and lifetime lubrication. The extra cost may be justified in situations where the cost of failure and replacement is high.

Pump efficiency can be increased with closer tolerances and through the use of backward curving, enclosed impeller vanes. Efficiency is not usually nearly as great a concern for smaller pumps as is manufacturing cost. Motor efficiency can be increased by substituting a brushless DC motor for a permanent-split capacitor motor or a permanent-split capacitor motor for a shaded-pole motor. If a great many pumps were to be ordered a manufacturer may consider offering a pump with a more energy efficient motor, making changes in the impeller and volute castings, or offering non-standard connection sizes. The market for small pumps for service line circulation is probably not large enough, however, to encourage such changes.

8.4 PLUMBING, MONITORING, CONTROLLING, AND ALARMING

Figure 8.9 shows the different ways circulation pumps can be plumbed. In Figure 8.9a, water to the customer is taken off downstream of the pump. If both the supply and return lines are the same size, the flow through the pump will be at its open discharge flow rate when water is taken off at twice the pumps open discharge flow rate. This is because at open discharge the pump contributes no head and both legs of the dual service line would have the same head loss and the same flow rate. A check valve can be added to allow water to bypass the pump when the withdrawal rate exceeds twice the pump open discharge flow rate as shown in Figure 8.9b. However, if this check valve were to fail open, there would be no circulation in the service line.

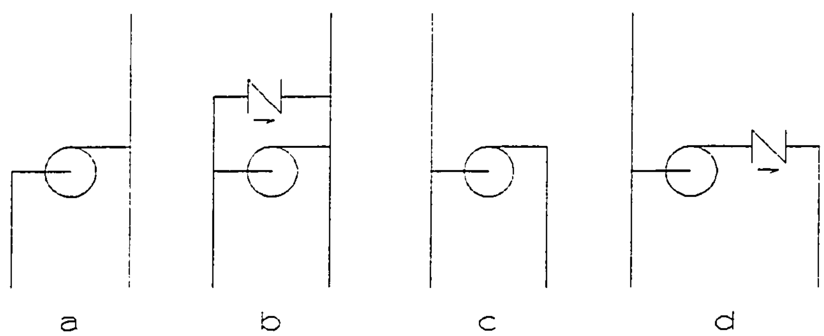


Figure 8.9: Pump Plumbing Arrangements

Figure 8.9c shows water taken off upstream of the pump. In this configuration flow through the pump may be stopped or reversed when water is withdrawn. The longer the service line, the smaller the diameter of the service line, or the lower the pump shut-off head, the easier it is for water to be drawn backwards through a small circulation pump. For example, a pump with a shut-off head of 10 kPa installed on dual lines 50 m long by 20 mm in diameter will be pumping at shut-off if 0.14 L/s [2.2 gpm] is being withdrawn and will have water drawn through it backwards at flows in excess of that as can be seen by referring to the system curve. With dual 100-m lines shut-off occurs at 0.10 L/s. A check valve can be added in series with the pump to prevent back flow as shown in Figure 8.9d. A check valve in series with a pump can substantially increase head loss and limit or even stop circulation in the line that would otherwise be present when the pump is not operating if pitorifices are used also, so it is not recommended in such cases.

The previous discussion assumed the dual service lines were the same size. If the return line in Figure 8.9a or the supply line in Figure 8.9c is substantially larger than the other line, then extremely large demand flows would be required to drop the pressure at the pump discharge or suction enough to cause flows in excess of open flow or flows going backwards through the pump. A large diameter supply line is often used where sprinkler systems are installed for fire protection. For sprinkler systems in single family residences, rather than using one large and one small service line, it may be advantageous to size both lines at some intermediate size, calculated to provide the same flow for the allowable pressure drop. If the pump fittings will not accommodate the line size, check valves may be advisable to avoid pressure drops at the reducers at the pump. The potential advantages of using an intermediate line size are decreased inventory requirements, increased practicality in using pitorifices for backup, and easier installation when using insulated conduits.

If water is forced backwards through a magnetic drive pump driven by a motor which has a higher torque than the magnetic coupling, the drive may decouple and the impeller assembly may vibrate and be damaged. Once decoupled, it may fail to recouple even after water flow has stopped. A check valve can be added to prevent decoupling as shown in Figure 8.9d. The brushless DC motor used by the Hartell HEL pump has a lower torque than the magnetic coupling and would not require a check valve.

All pumps are susceptible to air locks. Orientation of the shaft is critical in wet rotor pumps to prevent trapping air in the rotor chamber. Care should be taken to ensure proper orientation of the pump and to ensure air is not trapped in volutes or piping runs.

Pump speed control is an option on a few pumps. Generally, the lowest speed setting will still provide more than enough flow so it may be advisable to rewire the switch to ensure higher speeds will not be inadvertently used.

In the case of a brushless DC motor which can be operated at various voltages, speed control can be adjusted by selecting the appropriate DC power supply or adjusting the output of a variable output regulated power supply. An unregulated wall plug-in power supply rated to deliver the required amperage and voltage may be the most practical choice. If it is capable of delivering more amperage than required, its output voltage will be higher than its nominal rating but unless it is significantly oversized this should not be a problem.

Unlike an unregulated power supply, the output of a regulated supply will not drop if line voltage drops slightly. Regulated power supplies can be either linear or switching (Luchi, 1994). Linear-regulated power supplies can usually be expected to have efficiencies in the range of 25 to 50 percent or more. Switching-regulated power supplies can have efficiencies in the range of 65 to 85 percent or more and can be cheaper than linear-regulated power supplies even though they are more complex because a much smaller transformer is required. Regulated power supplies can offer some protection for the motor and can incorporate battery backup as an option, but they are more complex and costly than unregulated supplies. A custom DC power supply could incorporate circuitry that would monitor the impeller rotation of a magnetically coupled pump and signal an alarm if rotation speed was out of range.

Three power supplies were evaluated. The first was a specially ordered prototype unregulated supply with an output of 12 VDC at 1.25 A supplied by Jerome Industries Corp. (Elizabeth, New Jersey) with an estimated production cost of $28 each. The second was a specially ordered prototype linear-regulated power supply with a sealed 12 V, 4.0 A·hr lead-acid battery and monitoring lights supplied by MidAmerica Monitor Co. (Des Moines, Iowa) with an estimated production cost of $85. The last was an off-the-shelf switching regulator intended for a laptop computer which cost about $60.

The results of testing are shown in Table 8.2. The unregulated power supply gives acceptable efficiency and voltage stability over a fairly wide range of power output. The linear-regulated power supply has poor efficiency at the lower voltage outputs because regulation is achieved by essentially using a resistance voltage divider.

The linear-regulated power supply used a transformer with a rated efficiency of 80.2 percent, but it could be supplied with a special transformer with 90 to 95 percent efficiency for a higher production cost. The battery was switched out of the circuit during efficiency testing of this supply. Maintaining the charge on a sealed lead-acid battery can be done at a rate of C/500 amps where C is the A·hr rating

Table 8.2: DC Power Supply Test Results

Power Supply	W_{in}	W_{out}	A_{out}	V_{out}	η
Unregulated 12 VDC @ 1.25 A	8.4	5.6	0.41	13.8	0.68
	21.4	14.2	1.13	12.6	0.66
Linear-regulated 0 to 14 VDC @ 1.2 A	7.6	3.0	0.30	10	0.39
	12.4	5.0	0.50	10	0.40
	9.2	4.3	0.36	12	0.47
	22.4	11.1	0.93	12	0.50
	10.6	5.8	0.41	14	0.55
	20.5	11.8	0.84	14	0.57
Switching-regulated 9 VDC @ 1.1 A	5.7	2.8	0.29	9.68	0.50
	13.0	7.9	0.84	9.34	0.60

(Horowitz and Hill, 1989); this results in a maintenance charge rate of 8 mA which is not significant. Such a battery should last about 10 years.

A flow switch is sometimes used to switch on a heat trace or activate an alarm in the event of a loss of flow. A flow switch must be located upstream of the house supply line. A flow switch suitable for a service line will cost $100 to $600. The cheaper switches are paddle actuated and will generally require flow in the range of 0.3 to 0.7 L/s [5 to 10 gpm] in a service line and can be expected to result in pressure drops of 3 kPa [1 ft WC] or more. Magnetic or electronic flow switches cost far more than paddle switches but operate at flows down to 0.04 L/s and have negligible head loss. The energy savings that can be realized by using a lower head pump can justify the use of a more expensive flow switch. For example, the Grundfos UP15-42S pump performance curve and the Gems AFS-400 flow switch (Imo Industries Inc., Plainville, Connecticut) head loss curve are shown plotted from manufacturers' information in Figure 8.10. Also shown is the equivalent pump performance curve obtained by subtracting the flow switch losses from the original pump curve. (The concept of equivalent pump curves is discussed by Kanitz (1992).) Figure 8.11 shows the performance curve of a Grundfos model UM15-10B pump, the head loss curve of a McDonnell & Miller FS6 flow switch (Chicago, Illinois), and the resulting equivalent pump curve. Figure 8.12 shows the two equivalent pump curves with system curves for a 400 foot long, 1-inch diameter service and a 100 foot long, ¾-inch diameter service. For shorter service lines the resulting flow rate will be the same but the larger pump would cost $185 more a year to operate where electric costs are $0.42/kWh.

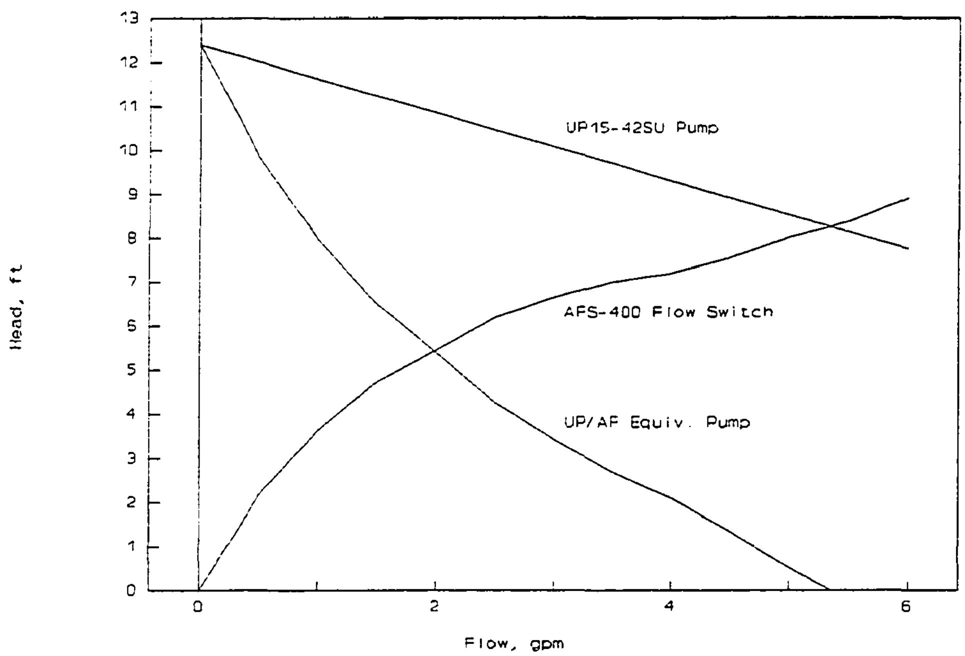


Figure 8.10: Grundfos UP15-42SU Pump with Gems Flow Switch

A temperature sensor on the surface of the pipe could conceivably be used as a flow/no-flow sensor if water temperature in the main is consistently and significantly lower than ambient temperature at the service take-off point. This should often be the case because a pump should be located in a heated space and water in the mains will usually be heated as little as possible. In the event of a loss of flow, the pipe surface in a heated space could warm sufficiently in 5 to 10 minutes to allow positive sensing without false alarms. An audible temperature alarm for about $20 is available from Extech Instruments (Waltham,

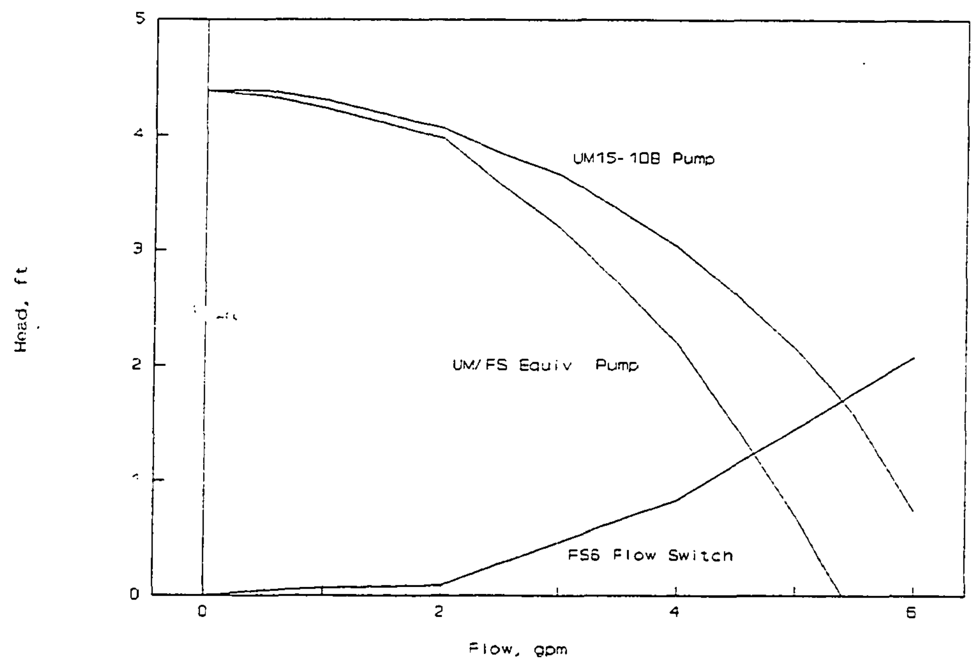


Figure 8.11: Grundfos UM15-10B Pump with McDonnell & Miller Flow Switch

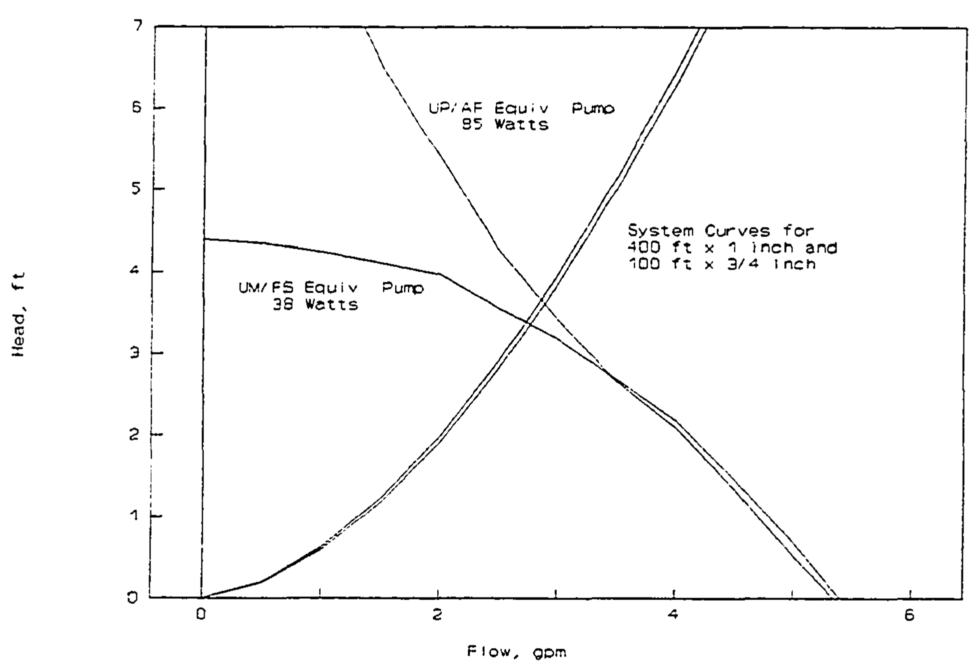


Figure 8.12: Equivalent Pump Performance and System Curves

Massachusetts) and controllers intended for controlling fans and refrigeration equipment using mechanically operated switches can be purchased for $50 and up from many other sources. Radio Shack (Fort Worth, Texas) sells a panel mounted battery powered thermometer/controller with a liquid crystal display for about $20 as part of a home alarm system. Such a device could be used to control a remote alarm.

Flow can also be monitored by the homeowner either directly or by observing the motor. Flow indicators with a visible flapper or rotating wheel could be used for a visual indication of flow, but the window may cloud with deposits over time. Motor operation is very easily checked by feeling for warmth or vibration, by listening, or by observing the shaft. Motor operation is usually a good indication of flow. A neon light mounted on an AC motor can be used to indicate the motor is switched on. An inexpensive sensor such as the model CR-20-G for about $20 manufactured by CR Magnetics Inc. (St. Louis, Missouri) can be used to monitor AC current to a motor. A pressure differential switch across a pump could also be used for alarming, but these switches cost more than $100. Proper motor operation or pump pressure differential is not always assurance of flow because a service line can freeze during a power outage with the pump restarting when power is restored without an alarm being indicated.

A prolonged power outage can result in freeze-up regardless of alarming and automatic heat trace back-up. Therefore, design should minimize the damage, cost, and inconvenience of a freeze-up. Strategies for accomplishing this include using soft drawn thick wall copper tubing installed to allow electric resistance thawing and using high density polyethylene tubing installed with heat trace in an insulated conduit. Access for internal thawing should be also considered. With these designs, it may not be cost effective to install alarms and automatic back-up systems.

It may be cost effective to use a thermostat bulb to turn a pump on automatically only when air or ground temperatures are near freezing; this could make a less expensive pump with higher operating costs more economical than a more efficient pump.

8.5 PUMP TEST RESULTS

8.5.1 Adding a Speed Controller to a Pump

Speed control devices intended for fans using shaded-pole or permanent-split capacitor motors cost $10 to $20 and could easily save $12/month or more where electric rates are $0.43/kWh, when used with existing, oversized circulation pumps. However, there are several potential problems with speed control. If the motor is started on the lower speed, the starting torque will also be less and the rotor may lock. Speed control may also result in higher rotor amperages although this is not likely to lead to an overheating problem, particularly for wet rotor pumps. Finally, speed control is one more item that may fail.

A Grundfos UP15-42SF was tested with two different models of speed controllers manufactured by Pass & Seymour (Syracuse, New York) designed for fan motors. One was a single-pole, 1.5 amp, 4-speed controller (Product No. 94004-1) which switched different capacitances in series with the motor and the other was a single-pole, 6 amp, variable speed controller (Product No. 94301-1) which uses solid state switching with silicon controlled rectifiers (Triac switched with Diac). The results are tabulated in Table 8.3 and demonstrate the possibility of substantial energy savings through the use of a relatively inexpensive speed controller.

Table 8.3: Speed Control Test Results for a Grundfos Model UP15-42S Pump

Control	Watts	Voltage		Shut-off Pressure kPa
		Peak	True RMS	
None	81	165	120	39*
5 μF	33	100	69	-**
9 μF	18	70	50	12
14 μF	5	29	29	3
Triac-Medium	25	142	57	12
Triac-Low	14	115	42	4

*From manufacturer; **Not measured

Switching a capacitor in line with an induction motor serves to lower the voltage without affecting the sinusoidal wave form. A lower voltage causes a lower torque which in turn causes lower speed due to increased slip. Shaded-pole motors have low starting torque and permanent-split capacitor motors have fairly low starting torque. Small fan motors generally use shaded-pole motors because they do not require very much starting torque. Centrifugal pumps are analogous to fans insofar as not requiring high starting torque, but a wet rotor pump with close tolerances between the rotor and shell may be susceptible to deposits which would prevent the rotor from starting when under reduced torque. A shaded-pole motor generally operates with more slip than a permanent-split capacitor motor so shaded-pole motors may be better candidates for speed control. Both fans and pumps have similar load curves and result in the speed climbing until limited by the rising torque requirement.

The Triac speed control also lowers the voltage but it does it by switching the current on and off so the wave form is only part of a sinusoidal form. Peak voltage using a Triac control is higher for the same speed but this did not result in a higher start-up torque. The lowest setting that could be obtained with the Triac control which would still allow restarting the motor had a peak voltage of 115 volts whereas with capacitor control the peak voltage was only 40 volts. The performances of the two controllers are roughly comparable; the Triac control was adjusted to give the same shut-off pressure as the 9μF case and required 25 watts compared to 18 watts for the capacitor control. The difference in wattage measurements may be due to instrument error by the watt · hour meter as a result of the distorted wave form due to the Triac firing mode, or it may be due to larger losses in the motor from the same effect. Both speed control devices show potential for use in retrofitting old or modifying new pumps to conserve energy.

Speed control is also possible with a step-down transformer which would reduce the voltage supplied to the pump. This was not tested but the costs would be about the same as the two speed controllers which were tested, and the results could be expected to be similar to those for the controller using the capacitors.

The starting torque for two Grundfos pumps was estimated by hanging weights on a light weight beam fastened to the impeller. The UP15-42 was fitted with the Grundfos 3-speed controller intended for the closed system model of the pump. Torques were 0.044, 0.079, and 0.133 N · m [450, 800, and 1,350 $gm_f \cdot cm$; 6.2, 11.1, and 18.7 $in \cdot oz_f$] for speed settings 1, 2 and 3 respectively. Speed setting 3 corresponds to the normal wiring for the open system UP15-42 without speed control. At this setting and with a 5 μF capacitor in series, the torque was 0.039 N · m [400 $gm_f \cdot cm$; 5.6 $in \cdot oz_f$]. The UM15-10 was found to have 0.118 N · m [1,200 $gm_f \cdot cm$; 16.7 $in \cdot oz_f$] of torque so it would be a better choice than a UP15-42 which uses 45 watts at low speed.

8.5.2 Pump Performance and Heat Loss Testing

Three pumps were selected for testing, one each from Grundfos, Little Giant, and Laing. Pump curves were obtained using a water-air manometer and a magnetic flow meter for differential pressure and flow measurement. Wattage was determined at each datum point using a watt · hour meter and a stopwatch. The pump inlets and outlets were connected to 0.5-m lengths of nominal ¾-inch diameter Type K copper water tube with piezometric rings located about 5 cm from the connection points. One-inch diameter reinforced vinyl hose was used to complete the loop and connect the flow meter. A five-gallon covered (but not sealed) tank was added to the loop to allow a total mass of about 20 kg of water to be circulated. Water temperature during testing was between 9 and 12°C and the air temperature was around 22°C. After taking data for the pump curve, the flow was set to 0.15 L/s and water and air temperatures were electronically logged at one-minute intervals as the water was allowed to warm over a 6 to 12 hour period. Tank and tubing were insulated and isolated from the floor to minimize heat transfer.

The data from the tests are shown in Figure 8.13 and given in Table 8.4. The pump curves determined for the Grundfos and Little Giant pumps are higher than those given by the manufacturers while that for the Laing pump is lower. The measured values for total wattage ($\dot{E}_t$) varied less than one watt throughout the range of flows (0 to 0.24 L/s) for all three pumps. The pumps were run dry to obtain an estimate for the power used by the motor alone ($\dot{E}_m$); this estimate probably errors on the high side because the unloaded motor will run faster and at a less efficient point. There was a one-watt drop in the power requirement when the pump was removed from the motor on the Little Giant pump which was the only pump which was coupled and not combined with the motor. This was presumed to be due to friction in the dry pump. The measured values for the motor with the pump running dry for the other two pumps were reduced by one watt to account for friction in the dry pumps. Viscous drag losses from the wet rotor or magnetic rotors when the pumps were pumping water were estimated to be about one watt or less and were considered to be part of the pump losses.

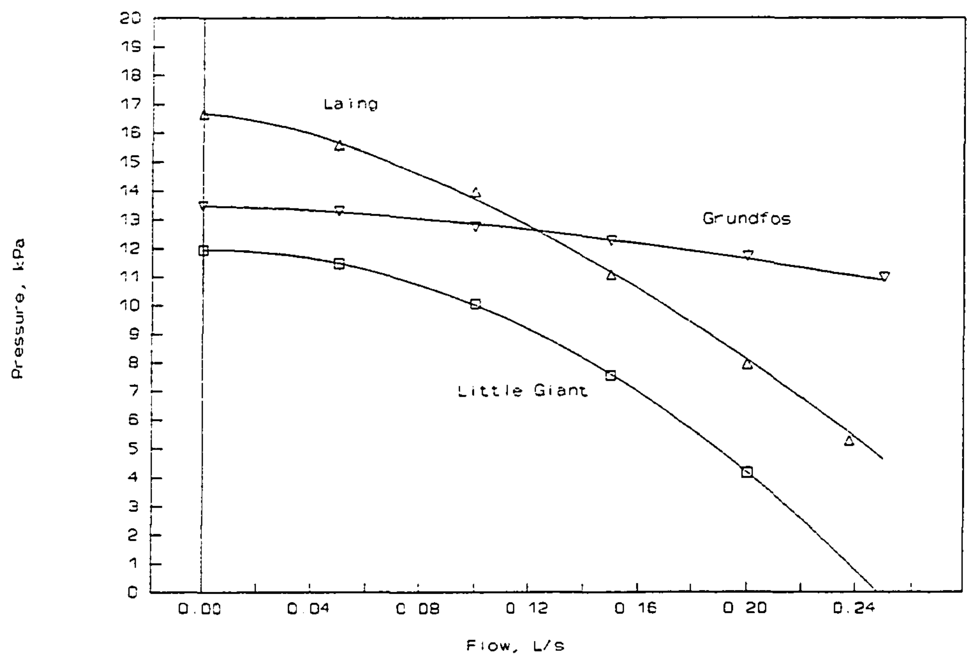


Figure 8.13: Pump Performance Test Data

Table 8.4: Pump Test Results at 0.15 L/s Flow

Pump Model	Man. Lit.		Measured			Calculated				
	ΔP	$\dot{E}_t$	ΔP	$\dot{E}_t$	$\dot{E}_m$	$\dot{E}_p$	$\dot{E}_w$	$\dot{E}_h$	η_m	η_p
Grundfos UM15-10B7	11.2	38	12.2	42	39	3	1.8	33	0.1	0.6
Laing SM-303	16.0	33	11.1	39	35	4	1.7	32	0.1	0.4
L. Giant CM6-100-3B	5.6	34	7.5	31	24	7	1.1	7	0.2	0.2

ΔP and $\dot{E}$ have units of kPa and Watts.
Subscripts t, m, p, w, and h refer to total, motor, pump, water, and heat.

The power used by the pumps alone was estimated by:

$$\dot{E}_p = \dot{E}_t - \dot{E}_m \tag{8.2}$$

where $\dot{E}$ refers the rate of energy use or the power and the subscripts p, t, and m refer to the pump, total, and motor requirements. The hydraulic power (also referred to as water power in the literature) which was delivered to the water was determined using the relationship:

$$\dot{E}_w = PQ \tag{8.3}$$

where P is the pressure differential across the pump and Q is the volumetric flow rate. Motor efficiency is given by:

$$\eta_m = \frac{\dot{E}_p}{\dot{E}_t} \tag{8.4}$$

The motor efficiency of 20 percent for the shaded-pole Little Giant motor agrees with a value of 23 percent for peak motor efficiency calculated from watt and torque curves for the motor which were supplied by the manufacturer. The low motor efficiencies calculated for the other pumps may be due to the air gap and stainless steel shell separating the wet rotor from the armature.

Pump efficiency is given by:

$$\eta_p = \frac{\dot{E}_w}{\dot{E}_p} \tag{8.5}$$

The low pump efficiency of the Little Giant pump is probably due to its straight vaned, open style impeller. These motor and pump efficiency values are very rough estimates but they do serve to give some insight into the nature of the pumps and motors.

The power which ultimately goes to heating the water ($\dot{E}_h$) is given by:

$$\dot{E}_h = \frac{mC_p dT_w}{dt} + \frac{(T_w - T_a)}{R} \tag{8.6}$$

where m is the mass of water in the loop, T_a is the ambient air temperature, R is the overall thermal resistance between the water and the air, and the other terms are as previously defined for Equation 8.1.

Figure 8.14 shows the water and ambient air temperatures recorded for the Laing pump. Taking the slope of the line for the water temperature at the intersection of the line for the air temperature ($T_w = T_a$) allows the second term of Equation 8.6 to be dropped and $\dot{E}_h$ to be directly calculated. Because the tank was not sealed, a small amount of heat was lost as water evaporated even when $T_w = T_a$. This heat loss rate was estimated at 1.4 watts by measuring the weight loss over a 24-hour period. This was added to the computed values for $\dot{E}_h$ to yield the values reported in Table 8.4. Covering the tank with a saturated cloth, putting oil on the surface of the water, or sealing the tank could have made this correction unnecessary.

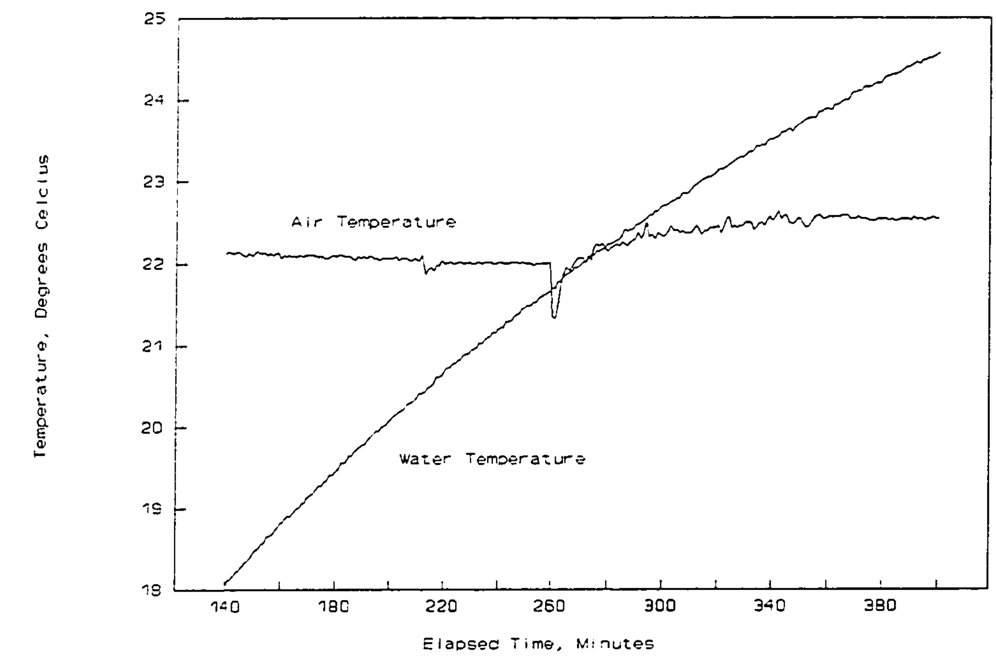


Figure 8.14: Water Warming Curve for Laing Pump

In the case of the Little Giant pump, so little heat was added to the water that it was more expedient to estimate a value of R from the curve for the Laing pump and calculate $\dot{E}_h$ using this value and the slope of the water temperature line at a point where $T_w < T_a$. Figure 8.15 shows the relative energy inputs and losses to air and water for the tested pumps. Both the Grundfos and Laing pumps have backward curving enclosed impellers while the Little Giant pump has straight, unenclosed radial vanes. A more efficient impeller or motor may be possible in the Little Giant pump.

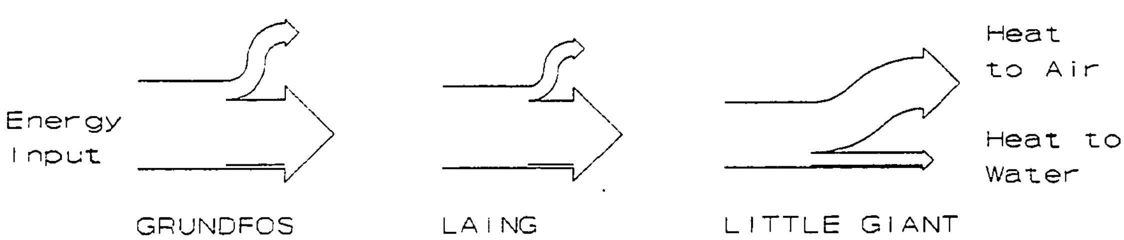


Figure 8.15: Relative Energy Inputs and Losses for Pumps at 0.15 L/s

8.5.3 Head Loss in Main Due to Circulation Pump

When a circulation pump is used with pitorifices it is possible to plumb it so that it opposes the flow in the main. Because the head generated from the pitorifice pair is generally only on the order of 0.5 kPa or 5 cm WC, the effect on the pumped flow rate in the service line can be expected to be negligible. A 6-inch diameter main with a 0.3 m/s [1.0 fps] flow velocity has a volumetric flow of 5.6 L/s [90 gpm] and the additional flow from the service line pump would be less the one-tenth that amount over a very short distance, so there should be little increase in the head loss in the main. The net loss due to the presence of ¾-inch BC&T pitorifices inserted to the center of a 6-inch diameter main with different service line circulation rates is shown in Figure 8.16. Pumping with the flow in the main helps reduce head loss but not by as much as pumping against the flow increases it. However, the effect in both cases is small and can be ignored.

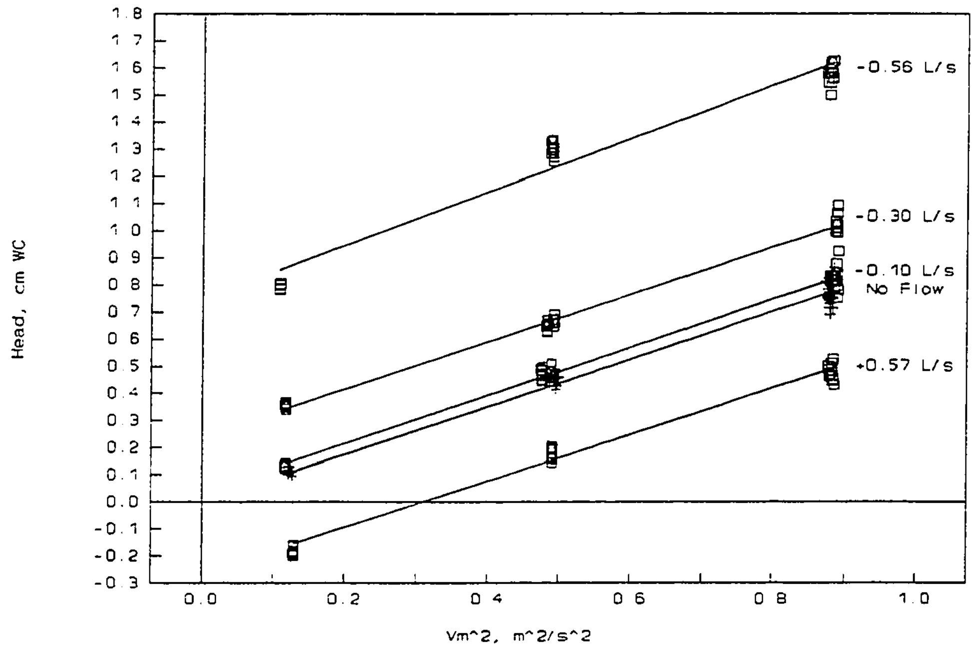


Figure 8.16: Net Head Loss in 6-inch Main with Pumping and ¾-inch BC&T Pitorifices

8.5.4 Operating Pumps Outside Their Flow Limits

Tests were conducted on the Grundfos UM15-10B7 to determine the effects of operating outside its performance curve. This could occur when water is withdrawn at a service as discussed in Section 8.4. The test results are shown in Table 8.5 with negative numbers for flow rates indicative of flow direction opposing the pump and a negative number for the speed indicative of reversal in direction. The rated current for the Grundfos UM15-10B7 is 0.4 amps at 115 volts and there was no evidence of vibration with flow outside the pump curve range. It is likely the pump would be safe regardless of the user tap location.

Table 8.5: Test Data for Grundfos Model UM15-10B7 Operated Outside of Pump Curve

Flow Rate (L/s)	Flow Rate (gpm)	Current Draw at 120 Volts (amps)	Rotation Speed (RPM)
-0.73	-11.5	0.405	1180
0	0	0.368	1550
0.81	12.8	0.380	1480
-0.50	-7.9	(turned off)	-3040
0.79	12.5	(turned off)	0

8.6 CONCLUSIONS

Most available pumps are inefficient and oversized for most service line freeze protection applications. Larger pumps may provide more starting and running torque which in turn may make them more dependable, but there is no evidence that this is so. A large, inefficient pump will add heat to the system, but this heating can be accomplished at less cost using waste heat or oil fueled boilers than by using electrical energy.

Use of smaller pumps is feasible based on analysis of available pumps and on experiments with speed controllers. If smaller pumps are used it is likely that flow switches and in-series check valves will not be practical because of the high expense of low head loss versions. Monitoring the temperature of the pipe loop inside the house may provide a workable alternative to a flow switch because a loss of flow will result in a rise in temperature. Also, if well insulated, heat traced, high density polyethylene service lines are used, the danger and cost of a freeze-up is limited so flow switches should not be needed. Finally, the energy savings from the smaller or more efficient pumps can be significant and justify higher first costs for these pumps.

Wet rotor pumps will transfer significant amounts of heat energy to the water but magnetic drive pumps may be more dependable where the pump is not operated for significant periods of time. Automatic control of pumps through ground or air temperature sensing switches may be cost effective in some situations.

Add-on speed controllers for existing, oversized pumps may offer substantial savings but long term dependability needs to be assured.

The use of small circulation pumps on certain services in a pitorifice system may allow cost savings in the system as a whole by allowing lower circulation rates in the main. Small circulation pumps may be a cost effective alternative to pitorifices in certain situations, particularly as efficiencies and the availability of smaller sizes improves.

CHAPTER 9: FIELD STUDIES

Field studies were performed to assess the performance of various water distribution systems and to supplement data collected in full scale testing. The studies are presented in the order in which they were initiated and conducted.

9.1 FAIRBANKS, 1466 CARR AVENUE

Many of the field test procedures were developed at the author's residence at 1466 Carr Avenue starting in April, 1992. The service line run from the main was traced by the Fairbanks Municipal Utilities Service (MUS) and found to be 72 feet [22 m] long. The service lines are ¾-inch diameter Type K copper water tube and tap a 12-inch diameter ductile iron water main. Pitorifices are probably the BC&T type with an OD of 0.75 inch and inserted about 6 inches apart. Flows in the main were 1.92 fps [0.585 m/s] based on the hydraulic analysis of the flow net model for the water system. This analysis was done using the computer program KYPIPE from the University of Kentucky, Lexington, Kentucky. The accuracy of this velocity was presumed to be ± 20 percent based on experience and spot field checks.

Figure 9.1 shows data collected using the flow test pipe described in Section 5.2.5. Data were collected during three separate runs made April 15 and 16, 1992 and corrected for losses in the flow test pipe. The line shown with x and y-axis intercepts of 0.059 m/s and 3.75 cm WC and a slope of $-0.64\ s^{-1}$ was determined by linear regression analysis. Flow is in the laminar region and the head available from the pitorifices can be assumed constant if the losses in pitorifice performance are assigned to equivalent length. This allows an equation for the line to be derived from Equations 3.3 and 3.4:

$$H = H_{SO} - \frac{64\,\nu\left(L_{eq} + L_T\right)}{2gd^2} V_S \tag{9.1}$$

where H is the head, H_{SO} is the shut-off head, ν is the kinematic viscosity of water (1.3×10^{-6} m^2/s at 10°C), L_{eq} is the total equivalent length of pitorifices and fittings, L_T is the total length of the service line, V_s is the velocity in the service line, g is the acceleration of gravity, and d is the ID of the service line. Using the slope of the line and the measured length of the service run, L_{eq} was found to be 10 m [33 ft]. This is in agreement with full scale test results of around 5 m for pitorifices and 2 m for the other fittings for a total of 7 m [23 ft]. The data could have been fitted with an exponential or quadratic curve if K_{ftg}s instead of L_{eq}s were used with K_{ftg} either taken as a constant or empirically related to V_s, but the precision of the data did not warrant this treatment.

While it is convenient to use flow velocities, it is really mass or volumetric flow rate per length

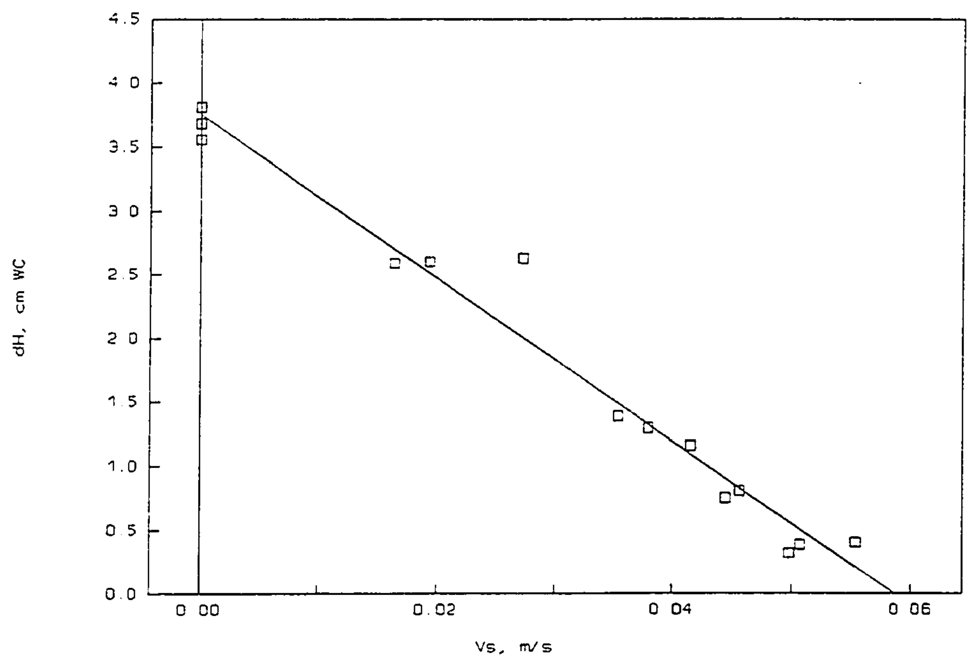


Figure 9.1: Service Line Flow Measurement with Flow Test Pipe

of run which is significant. A velocity of V_s=0.059 m/s [0.19 fps] in nominal ¾-inch diameter Type K copper water tube corresponds to a volumetric flow rate of Q=0.017 L/s [0.26 gpm]. The volumetric flow rate per length of run for the service is therefore 0.076 L/s/100 m of run [0.37 gpm/100 ft of run]. Because this service has never been reported to have frozen, it is reasonable to assume that other services with similar heat loss rates and the same volumetric flow rates per length of run would not freeze either with the same temperature for water in the main.

The velocity in the main can be estimated by rearranging Equation 4.14 and noting that $H_{SO}=H_{po}$:

$$V_m = \sqrt{\frac{2gH_{SO}}{K_{po}}} \tag{9.2}$$

Assuming $K_{po}=2$ and using a shut-off head of 3.75 cm WC, the flow in the main is predicted to be 0.6 m/s [2.0 fps] agreeing closely with the computer model.

Figure 9.2 shows temperature data collected using the flow test pipe on April 25, 1992. Circulation was stopped, then restarted 60 minutes later. The heating rate per unit length of line exposed to the indoor temperatures is:

$$\frac{\dot{E}_h}{L} = \frac{\Sigma\left(mC_p\right)}{L}\frac{dT}{dt} = \Sigma\left(\rho A C_p\right)\frac{dT}{dt} \tag{9.3}$$

where m is mass, ρ is density, C_p is heat capacity, dT/dt is the rate of change of the temperature. This assumes both the water and the copper warm at the same rate and are always at the same temperature. The summation, therefore, is required to include both the water and the copper. The water initially warms at a rate of 0.25 C°/min. Using ρA=0.28 kg/m and C_p=4,200 J/kg · C° for water and ρA=1.98 kg/m and C_p=420 J/kg · C° for the copper line with the mass of the tee fitting included. (Copper has a specific gravity of 8.8.) These result in $\Sigma\rho A C_p$=2,000 J/m · C° and an initial heating rate of 8.4 W/m.

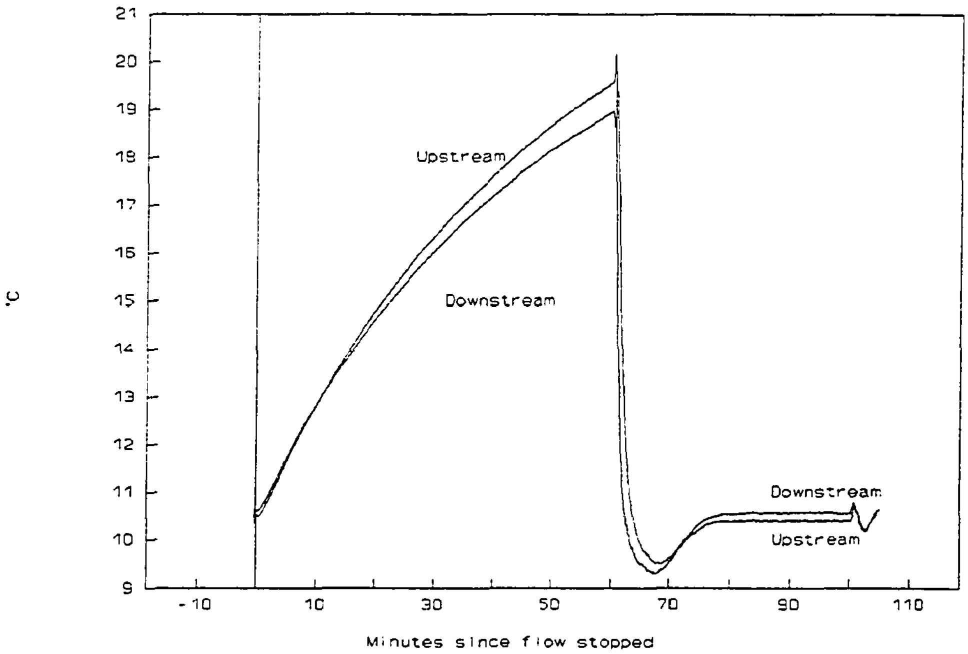


Figure 9.2: Service Line Temperatures with Flow Stopped and Restarted

Equation 9.3 can also be written as:

$$\frac{\dot{E}_h}{L} = \pi D h \left(T_A - T\right) = \sum \left(\rho A C_p\right)\frac{dT}{dt} \tag{9.4}$$

where D is the outside diameter of the pipe, h is the overall heat transfer coefficient, and T_A is the ambient air temperature. Using the initial heating rate of 8.4 W/m, h is found to be about 10 W/m²C°. Equation 9.4 can be integrated and the warming curve fit with an exponential equation. The time constant in the exponential will also yield a value for h.

The time required to reestablish laminar flow is given by (Letelier, 1976):

$$t = \frac{0.1729 r^2}{\nu} \ln\left[\frac{V_0}{V_0 - V}\right] \tag{9.5}$$

where r is the inside radius or the line, V_0 is the velocity at $t=\infty$, and V is the instantaneous velocity. To reach $V=0.95V_0$ takes only 35 seconds. Several minutes after flow was reestablished, a slug of water about 1 C° colder than normal passes. For the copper water tube service without a fitting, $\Sigma\rho AC_p=1{,}570$ J/m·C°. The apparent cooling rate of about 1 C°/hr in the buried service line for an equivalent heat loss rate per unit length of buried line of around 0.4 W/m assuming the thermal heat capacity of the insulation is negligible and the temperature drop of the water is small compared to the temperature differential across the insulation.

Service lines are usually four to five feet deep. Ground temperatures have been recorded down to 15°F at depths of six feet but minimums of 25°F at depths of four feet are more common (Page et al., 1957). MUS specifications call for a 3-inch thickness of urethane foam with a thermal conductivity of $k_I=0.017$ Btu/hr·ft·F° [0.029 W/m·K°] on service lines. The average thermal conductivity of insulation buried up to 14 years has been measured to be $k_I=0.015$ Btu/hr·ft·F° [0.026 W/m·K°] (Haigh, 1986).

Assuming insulation equivalent to an annulus with inside and outside diameters d=2 inches and D=8 inches and insulation with $k_I=0.026$ W/m·K°, the thermal resistance per unit length is:

$$R = \frac{\ln(D/d)}{2\pi k_I} = \frac{\ln(8/2)}{(2)(3.14)(0.026\,W/m\cdot C^\circ)} = 8.5\,m\cdot C^\circ/W \qquad (9.6)$$

Assuming an average ground temperature of $T_G=-4$°C [25°F] at the service line insulation and an average water temperature of $T_W=10$°C, the steady state heat loss rate per meter should be:

$$\dot{E}/L = \frac{T_W-T_G}{R} = \frac{14\,C^\circ}{8.5\,m\cdot C^\circ/m} = 1.6\,W/m \qquad (9.7)$$

The temperature of water in the service line after a time t=60 minutes assuming negligible thermal mass of the service line and insulation would be:

$$T = T_G+(T_W-T_G)\exp\left[\frac{-t}{R\sum\rho AC_p}\right] = -4+14\exp\left[\frac{(-3600\,s)(J/W\cdot s)}{(8.5\,m\cdot C^\circ/W)(1570\,J/m\cdot C^\circ)}\right] = 7\,^\circ C \qquad (9.8)$$

Both this estimated temperature and the absence of a flat portion at the minimum suggests that the actual cooling rate may be greater but that some damping of the temperature is occurring.

The downstream minimum for water temperature lags the upstream minimum by about 50 seconds. Because the thermistors were 1.4 m apart this suggests $V_s=0.03$ m/s, one-half of what was determined using the head loss curve. The discrepancy may be due to very small differences in placement of the thermistors in the flow velocity profile. Flow is laminar and the velocity profile is fairly great, with flow in the center being twice the average velocity.

The spike and dip at the end of the temperature profile resulted from withdrawing water from a faucet for about a minute at a rate of about 0.35 L/s [5.5 gpm] which represents about a 0.6 m/s [2 fps] flow in each leg of the service line loop for a transit time from the main of 35 seconds during which the temperature drop would be insignificant. The temperature climbs at first due to flow reversal in the test pipe section; the tee to the house is off the upstream leg so when there is withdrawal, the water flow past the temperature sensors reverses and the water that has been warmed in the room for the longest time flows back past the thermistors again. The temperature then drops to a low value which represents the temperature of water in the main. The temperature climbs again after withdrawal is stopped. The close agreement between the thermistors shows they have the same temperature response.

Assuming a cool-down rate of 1 C°/hr, and a 6 minute travel time, the temperature in the service line should cool about 0.1 C°. This is approximately the difference between the upstream thermistor reading and the reading taken when water is being withdrawn, allowing for some heating at the house.

Additional testing was done starting at 17:50 on Saturday, November 27 and continued until Tuesday, November 30, 1993 using instrumentation for flow, pressure, and temperature monitoring and logging every 2 minutes. Before connecting this equipment, the bypass manometer described in Section 5.2.5 was used. The shut-off head was 3.2 to 4.0 cm WC and the bypass head was 0.8 cm WC. This yields an estimated minimum flow rate estimate of 0.013 L/s [0.2 gpm] or 0.045 m/s in the service and a main velocity of about 0.6 m/s [2.0 fps]. Figure 9.3 shows the flow rate stays fairly constant and is equivalent to V_s=0.05 m/s. The flow drops when very minor withdrawal rates occur at the house and reverses for larger withdrawal rates. Flow was throttled to half the normal rate at the conclusion of testing in an attempt to determine cool-down rates, but the temperature differences were too small to allow meaningful calculations.

Figures 9.4 and 9.5 show the pressure and temperature readings. The pressure stays fairly constant but temperature trends downward. Temperature does not vary with service line flow rate but follows the actual temperature variations in the water system water due to relative make-up rates and temperature of make-up water. Sharp peaks about 0.4 C° high in the temperature presumably represent instances where water that has warmed in the service is withdrawn backwards back past the thermistor again as water is withdrawn at a tap in the house.

The service line length was estimated by shutting off one leg and noting the pressure drop while withdrawing water at different rates. Figure 9.6 shows the pressure with no withdrawal and for withdrawal rates of 5.4 and 13.8 gpm [0.34 and 0.87 L/s]. The pressures logged while withdrawal was adjusted are omitted for clarity. A line determined by a linear regression analysis is shown for the no-withdrawal case. The actual pressure readings were subtracted from those of the linear fit for the no-withdrawal case and averaged to find drops of 4.0 and 23.5 psi. By assuming a total fitting loss factor of K_{ftgs}=5 and a

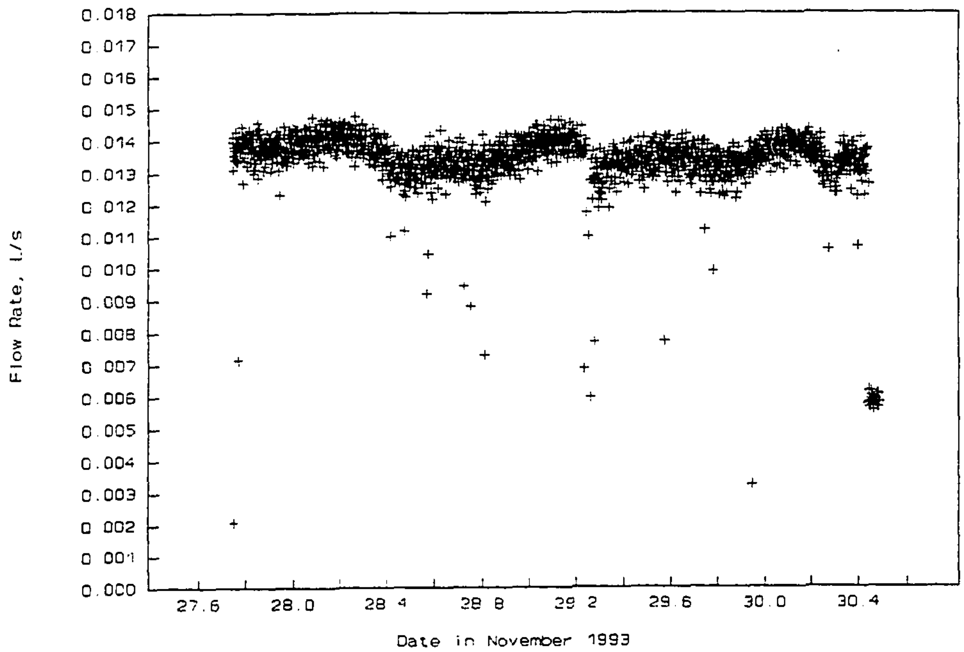


Figure 9.3: Service Line Flow Rate Measurements

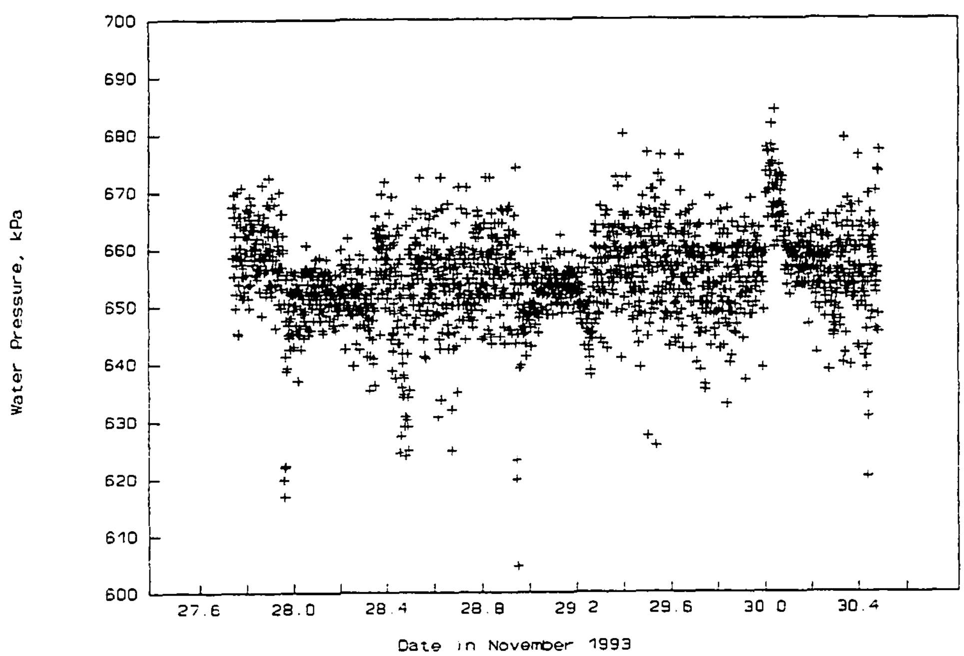


Figure 9.4: Service Line Pressure Measurements

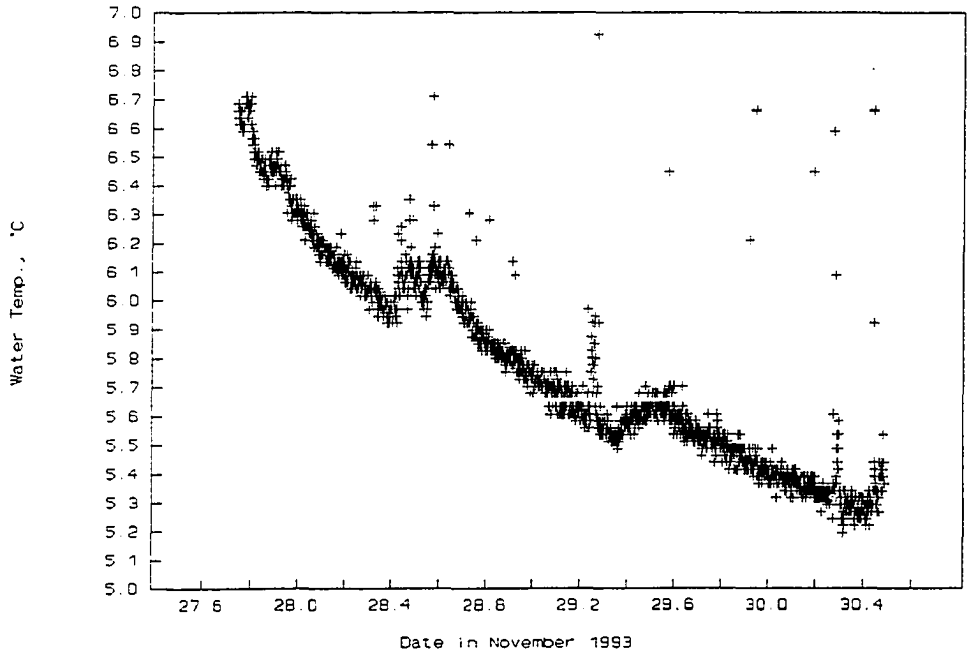


Figure 9.5: Service Line Temperature Measurements

hydraulic roughness of ϵ=0.000005 ft [0.0015 mm] for drawn tubing (Moody, 1944) the length of one leg of the service line was found using Equations 4.16 and 4.18 to be 23.1 and 25.5 m [76 and 84 ft] for the two withdrawal rates.

The service line run length was also estimated from the oscillation rate as described in Section 5.5. The oscillation rate was found to be 12 $\pm$1 Hz. A service run distance of 33 m [108 ft] is found by dividing by two and subtracting the length of the vinyl hose used. Due to the difficulties and inaccuracies of both the pressure drop and oscillation rate methods of estimating line length, neither method was used in subsequent investigations.

9.2 MARSHALL

The city of Marshall, Alaska was visited August 12, 1992 for two days to identify problems with services that had frozen the previous winter and to determine minimum flows in those services with no freezing problems. Three separate instruments for flow determination were tried: a Clorius magnetic flow meter borrowed from MUS, the flow test pipe, and the first version of the bypass manometer. The magnetic flow meter was not used in any field measurements because it was malfunctioning. The bypass manometer proved entirely satisfactory and it was far easier to use than the flow test pipe. U.S. Public Health Service (PHS) engineers Jim Magnuson and Scott Matthews assisted at the beginning and the city water plant operator David Fitka assisted throughout the investigation.

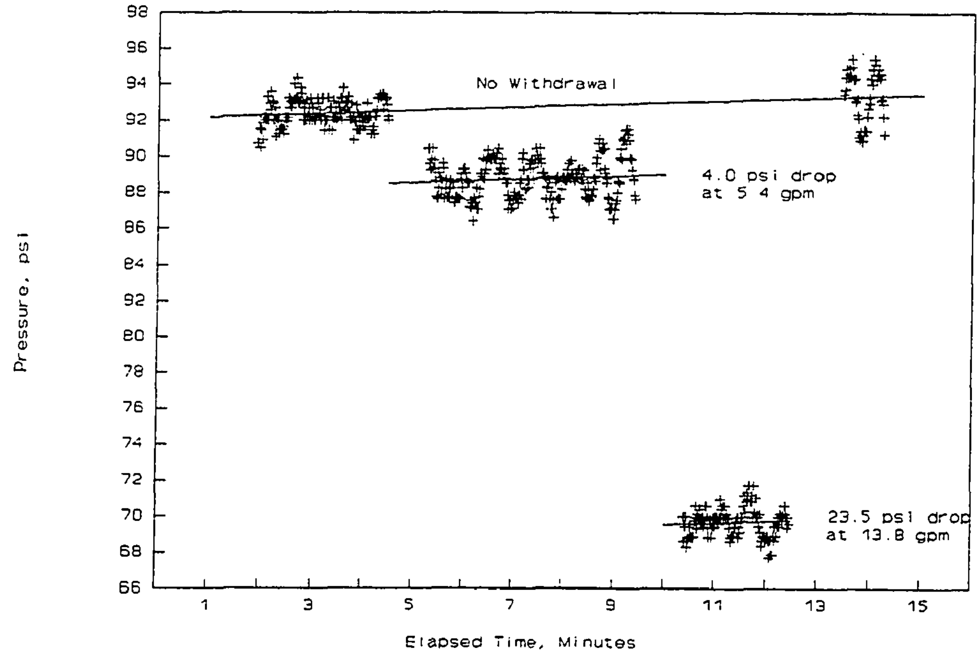


Figure 9.6: Using Pressure Drop to Estimate Service Line Length

9.2.1 Pumphouse Plumbing

Water is circulated year round. The existing water meter on the looped main could not be used in the study because it was found to be inaccurate and could not be repaired or calibrated with the equipment on hand. A McCrometer turbine meter on the return line at Marshall appeared to be a 3-inch model and was plumbed immediately after a 2½-inch bend which is poor practice for accuracy and meter life. The meter was being bypassed because it was making a rasping noise. It was put back on line and registered 60 to 65 gpm with one pump operating and 85 to 90 gpm with two pumps operating. Pitorifice shut-off heads measured at a service were 2.2 and 3.0 inches for these two cases. Assuming a main with an ID of 4.0 inches and an IBC&T type pitorifice with an OD of 0.9375 inch yields an estimate of K_{po}=2.6. The flow velocities in the main should therefore be 2.1 and 2.5 fps [0.64 and 0.76 m/s] for the two cases and the volumetric flow rates should be about 85 and 100 gpm respectively. This agrees well with the design flow of about 90 gpm with one pump operating. Withdrawing an estimated 50 gpm downstream of the meter did not result in any flow being indicated by the meter; unfortunately larger flows could not be withdrawn through the 1-inch tap provided and the meter could not be calibrated.

All field tests described below were made with only one circulation pump on. Loop return temperature was 48°F [9°C].

9.2.2 Service Descriptions

Four services which experienced freeze-ups during the previous winter and three which had no history of problems were inspected. Freezing problems seemed to be due to: (1) poor insulation which allowed air infiltration, (2) power loss resulting in loss of circulation in the main, or (3) air locks in the service loop plumbing. Low circulation rates did not seem to be a problem.

Figure 9.7 shows schematics of the three types of services present in Marshall. In the 1974 design ¾-inch diameter copper tube was brought up through the floor of the house to angle stop valves and the house plumbing was supplied from a ¾ × ½-inch tee through a union and gate valve. The 1978 design was similar except that gate valves were used instead of angle valves, two dielectric unions were used, and the house plumbing was ¾-inch diameter. In the 1983 design 1-inch diameter high density polyethylene (HDPE) pipe was run to a box attached to the outside of the house. The pipe was terminated in angle plug valves and these were attached with 1-inch diameter flexible reinforced vinyl hoses to ½-inch diameter copper lines which ran through the wall into the house and then looped back out to the box.

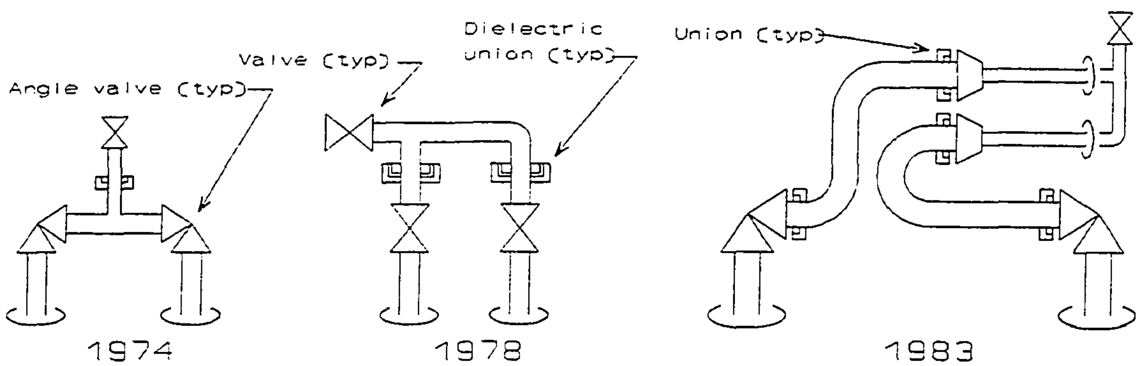


Figure 9.7: Service Connection Schematics for Marshall

A total of 29 active service connections dating from 1974 and 1978 were identified in the field. Another 15 services dating from 1983 were identified on PHS Project AN-83-655 Aerial Photo Sheet No. D-2. All four service line freeze-ups experienced during the winter of 1991-1992 were in these newer services.

All connections are served by the same loop and all have pitorifices. Most services are heat traced. One service was installed in 1990, and it is the only service which has an individual circulation pump. It was not feasible to open and test any of the older service lines or the one with the individual circulating pump because of the construction of the boxes.

9.2.3 Frozen Services

9.2.3.1 1984 BIA House Number 65

The owner reported her service line froze despite her use of the heat trace during very cold weather. The operator reported that when he was notified of the freeze-up he found the heat trace was inoperable and replaced it.

The service run is about 6 feet horizontal then 6 feet vertical. One-inch diameter HDPE lines run from pitorifices in the main inside 4-inch diameter arctic carrier pipe (insulated utiliduct) to an insulated box on the side of the house. The HDPE lines terminate in 1-inch Ford angle stop plug valves which were connected to 3-foot lengths of 1-inch diameter reinforced vinyl hose. The vinyl hose was connected to ½-inch diameter copper tube which goes into the house to the service take-off point and then returns to the box. The total length of the ½-inch diameter copper loop was 5 feet and there were six elbows and one tee in the flow path. Heat tracing was wrapped around the copper tubing and hoses in the box and was shoved down the arctic carrier pipe. The box was stuffed with fiberglass batt insulation.

In preparation for the testing, the vinyl hoses were disconnected from the ½-inch diameter copper water tube and the service lines flushed from the main. Water was initially rusty colored but quickly cleared. A bypass manometer was inserted between the hoses and a shut-off head of 7 cm and a bypass head of 5.5 cm were measured. The bypass head was off the scale for the calibration curve but a lower limit of 1.5 gpm could be assumed and a flow of 2.0 gpm [0.13 L/s] was estimated. This indicated there was no problem with the pitorifices or the service line to the house. The total equivalent length of pitorifices and fittings was not computed because there was uncertainty about the actual induced flow rate and because the flows were estimated to be in the turbulent flow range of $3{,}000 < Re < 4{,}000$.

The flow test pipe and the house plumbing were then connected into the service line loop and the normal flow in the service was found to be about 0.04 L/s [0.63 gpm]. This is much lower than it would be if the house were plumbed with ¾ or 1-inch diameter copper tube for the loop in the house, but it is still more than adequate to prevent freezing in a properly insulated system. Although flow may have been laminar in the 1-inch diameter line, it was turbulent ($Re \approx 3{,}000$) in the ½-inch diameter line. Because of this and the short run length, the equivalent length for the pitorifices and fittings was not calculated.

A closer inspection of the box was then made. Gaps 2 to 3 inches wide were noticed between the house and the box and between the box and the carrier pipe. These permitted cold air to flow unimpeded against the plumbing in the box. These gaps appeared to be the result of differential movement between the house and the carrier pipe. Had the box and connection been properly constructed and installed, these gaps would not have occurred. Figure 9.8 shows the installation. This was by far the most poorly constructed service box visited. Some of the other service boxes had gaps but all other service connections of this type had rubber flexible boots to prevent air leaks between the box and carrier pipe.

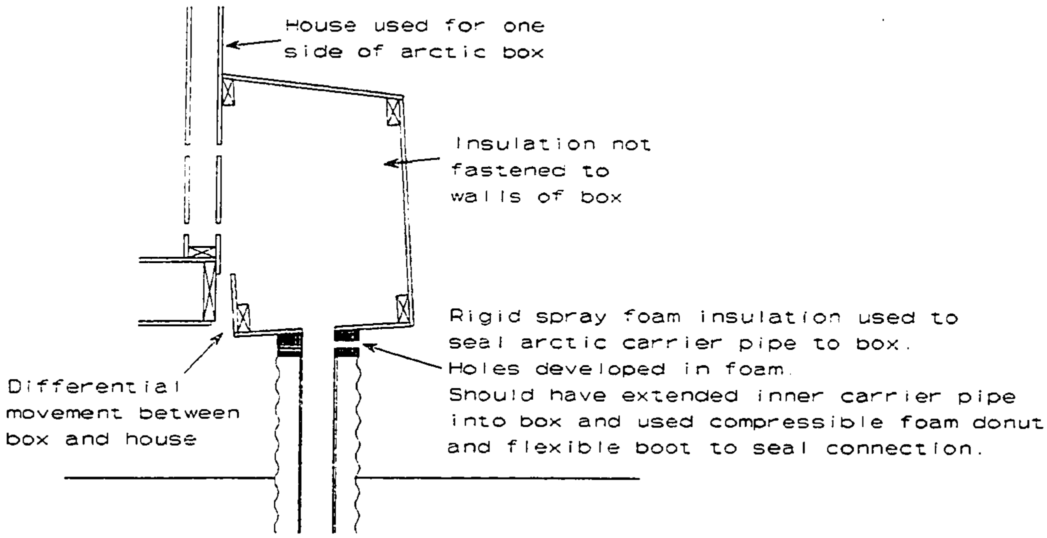


Figure 9.8: Poorly Constructed Service Connection Box

Air bubbles were observed forming in water that was freshly withdrawn from the service. This suggested that flow in the service may have stopped due to an air lock. With a deep well, a hydromatic tank, and low system pressure, the Marshall system may be susceptible to air in the lines (see Appendix E). If enough air found its way into the service, circulation could have been stopped long enough for infiltrating cold air to cause a freeze-up.

9.2.3.2 City Health Clinic Building

The service to this building is identical to those of the 1984 HUD homes. The flow was estimated to be 0.9 gpm based on shut-off and bypass heads of 7 and 4 cm WC. Loss of circulation due to a power failure or an air lock was suggested.

9.2.3.3 1984 BIA House Number 79

This house is about 270 feet from the main and was first served in 1990. It is the only house in Marshall with a circulation pump. The owner said he had left the house unoccupied for a month with the heat trace plugged in to prevent freezing of his water line. When he returned, the line was frozen and his monthly electric bill was $200 more than usual. The operator believed that the circuit breaker tripped due to power supply problems and that once the circulation pump and heat trace were shut off, the line froze. It is also possible that only the line into the house froze. The total length of 5-watt/ft self limiting heat trace used was reportedly over 300 feet. Circulating water keeps the interior of the carrier pipe cool and it is reasonable to expect the output of the tape may have been in excess of one watt/ft. At one watt/ft and with an electric rate of $0.42/Kwh, the monthly charge would be about $90.

The circulation pump was located in the service box and water was not circulated into the house as it was done for all other services in Marshall. Figure 9.9 shows a schematic of the plumbing in the box. This installation could be improved by having the pump circulate water through the looped house plumbing or by adding a second heat trace just for the house plumbing. The pump should be located inside the house where its operation could be more easily monitored by the occupants.

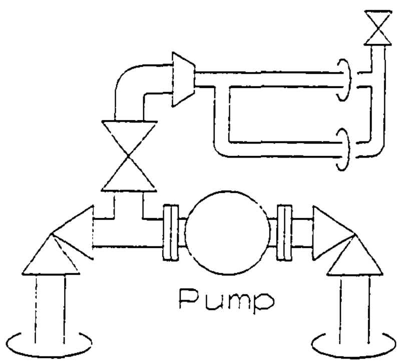


Figure 9.9: Poor Plumbing of Pump for Service

9.2.3.4 1983 BIA House Number 43

The operator said the owner shut off the service line valves because of a leak in the house which was caused by freezing. He was not sure where the leak in the house plumbing was, it could have been in the service loop or downstream of the supply tee. The owner was not available so the cause was not resolved. The service box was inconveniently high so the flow in this line was not tested.

9.2.4 Other Services Investigated

9.2.4.1 1974 BIA House Number 26

This service consists of ¾-inch diameter copper lines coming up from under the house into the laundry area with the tap on the highest point, typical of the other houses built in 1974. According to the occupants, the service has never frozen. The heat trace is manually activated when it gets extremely cold outside.

9.2.4.2 1978 HUD House Number 6

This service consists of ¾-inch diameter copper lines coming up from under the house into the laundry area with the tap on the highest point. The lines up to and past the unions were insulated with urethane foam. It is typical of the other 23 houses built in 1978 which also did not freeze.

9.2.4.3 1984 BIA House Number 71

This service is similar to six others, four of which froze that past winter. The service box is located an estimated 56 feet [17 m] from the main and this is by far the longest service line in Marshall without a circulating pump. The service loop in the house consists of about 18 feet of ½-inch diameter copper tube and it contained six elbows. The 112 ft of 1-inch diameter HDPE and 18 ft of ½-inch diameter copper tube is equivalent to a single 100-ft length of ¾-inch diameter copper tube assuming laminar flow and not counting fittings and using Equation 7.2. There was no heat trace and there was no insulation in the box. This service has never frozen according to the occupants.

The induced circulation flow was estimated to be about 0.13 gpm [0.0082 L/s]. This yields 0.23 gpm/100 ft of run [0.048 L/s/100 m of run] compared to 0.37 gpm/100 ft of run for 1466 Carr, Fairbanks. One of the faucets in the house was later observed to be leaking at an estimated rate of a little more than 0.1 gpm and this probably affected the measurement.

9.2.5 Summary of Findings at Marshall

Residents seemed to rely on heat tracing to prevent freezing. This should be unnecessary with a properly functioning system but may be the result of some problems in the past. Operating heat trace may be prudent for the copper services because these cannot withstand as many freeze-thaw cycles as HDPE services. The use of HDPE service lines with self limiting heat trace allows services to freeze without significant consequences; residents with these should use the heat tracing only for thawing purposes. When self limiting heat trace is turned on and water is withdrawn or circulated, the cold water will keep the heat trace activated and energy will be wasted heating the water.

Heat tracing for freeze protection should not be wrapped directly on the line, particularly if it is a copper line with circulating water. The pipe should be insulated and the heat trace used to keep the space around the pipe warm. A lower wattage tape can be used for freeze protection than for thawing. Unlike a thawing heat tape which needs to be sized to thaw a line within a short period of time, a freeze protection tape just needs to balance the worst case heat losses. If freeze protection heating is done, it may be advisable to use a thermostat for control and to install a low temperature alarm.

Loose insulation tends to get removed or degraded over time. Instead of filling the box with insulation, the pipe in the box and the walls of the box should be insulated separately. Sealing the box against air infiltration and accommodating movement are very important. Consideration should also be given to enlarging the opening between the box and the house so that the interior of the box is kept warm.

Some thought should be given to limiting the possibility of air locks when designing service runs, see Appendix E. Service loops should not include small diameter (< ¾-inch) piping because it can severely limit flows.

The following checklist for site visits was developed as a result of the Marshall study:

Any leaky faucets?

Is water deliberately run to prevent freezing?

What is history of freeze-ups?

Are there one or more heat trace lines?

How often, how long, when is heat trace turned on?

How many occupants?

Is house unoccupied for long periods?

Is house cold or drafty?

Any problems with air in the lines?

Does air ever come out when faucets are opened?

Does the loop extend into the house?

Where is the service tap in relation to the loop high point?

Any spots where outside air can infiltrate against plumbing?

Is the insulation good?

9.3 LOWER KALSKAG

The city of Lower Kalskag, Alaska was visited August 14, 1992, for two days for the same purpose as Marshall and the same equipment was used. PHS engineer Tim Edwards, PHS foreman Frank Turner, and the city water plant operator Hank Aloysius assisted in the study.

Each of the three loops serving Lower Kalskag have a McCrometer turbine meter on the return leg. None of these meters were working. The operator reported that he found the cable broken on one meter and suggested that might be the problem with the others.

An attempt was made to measure flows in the mains and services using the bypass manometer. No differential head was detected across services in loops one and two, indicative of a flow in the main under 10 gpm. A shut-off head of about 3 mm was noted in a service in loop three, yielding an estimated average velocity in the main of V_m=0.15 m/s [0.5 fps] assuming K_{po}=2.6. For a 4-inch diameter main this is a flow rate of about 20 gpm.

The fire hydrant on loop three was opened and the main flushed alternately from both the supply and return sides. It was then noticed that all the loops had basket strainers. The fine mesh strainer baskets on loops one and three were found to be plugged with a reddish mass of slimy small flakes of material, presumably iron deposits sloughed off the PVC pipe walls. The strainer basket on loop two was fairly clean. All strainer baskets were cleaned and those for loops one and two were replaced. The operator said he had never opened these strainers in the four years he had been operating the plant, but that he had operated the blow-off valves and thought this was all that was needed to keep the screens clean. Unfortunately, these strainers were plumbed with ¾-inch ball valves for blow-off but 1½ or 1¼-inch valve should have been used.

With the basket out of the strainer for loop three and with both circulation pumps running, the shut-off and bypass heads in the first service on the loop were found to be 5.5 and 3.5 cm. These led to estimates of V_m=0.64 m/s [2.1 fps], Q_m=80 gpm, and V_s=0.2 gpm [0.013 L/s]. The 1-inch diameter HDPE service line had a horizontal run of 70 feet [21.3 m] and an estimated vertical rise to the box of 8 feet where it terminated at Ford angle stop valves. The valves transitioned through unions and short pieces of HDPE line to approximately 2 feet of 1-inch diameter copper tube with an elbow and a tee inside the house. Solving Equation 9.1 for L_{eq} with H=0, corresponding to unobstructed flow:

$$L_{eq} = \frac{2gd^2H_{SO}}{64\nu V_S} - L_T = \frac{\pi d^4 g H_{SO}}{128\nu Q} - L_T \qquad (9.9)$$

Using L_T=42.7 m, d=0.0273 m [1.076 inches], and $\nu=1.3\times10^{-6}$ m²/s, L_{eq}=390 m. This exceptionally high value for L_{eq} suggests a damaged or partially plugged pitorifice or line, or a line size less than 1-inch diameter.

Pump B on loop three was then pulled and the impeller inspected. It was stained reddish brown and was slightly plugged but was not noticeably worn or corroded. After returning the pump to service, its shut-off head was found to agree with that of pump A. The individual pumps on the other loops also had shut-off heads which agreed closely with one another. It was presumed that all the pump impellers were relatively free of debris.

The screen was replaced in the strainer on loop three and the service tested again. The results were inconclusive. Flow in the main seemed to fluctuate between 40 and 55 gpm and flow in the service was less than 0.1 gpm regardless of whether one or two pumps were operated. Because the hydrant was not flushed prior to testing, the low, fluctuating flow may have been due to problems with air in the line. The low flow may have also been due to a high head loss across the strainer basket.

The experience at Lower Kalskag serves to emphasize that strainers should not be used except right after work has been done on the mains and debris needs to be flushed out. Also, orifice plates or flow nozzles should be considered for monitoring return flow rates because turbine meters often malfunction.

9.4 FAIRBANKS, COLLEGE UTILITIES CORPORATION (CUC) SERVICE AREA

A total of nine residences were visited between January 6 and March 12, 1993 in the Fairbanks CUC service area with Mike Lester of CUC assisting. All residents were asked a set of questions developed as a result of the Marshall trip. No freezing problems, water wasting, air in lines, heat trace, or leaky fixtures were reported. The services were all installed between 1978 and 1988, household demands ranged from 55 to 157 gpd averaged over the 1992 calendar year, and water temperatures were 4 to 5°C [39 to 41°F].

Flow tests were not possible at four of the nine residences visited. Two residences had circulation pumps; one of these had a service run of over 100 feet and the other had an inoperable pump and shut-off valves that would not seal. The other two residences had pack joints instead of unions and these would have been too difficult to replace. The site with the inoperable circulation pump (Grundfos Model UP25-42SF) had a service distance of only 55 feet so it was suspected that there would be adequate flow in spite of the pump. To confirm this, the hydraulic resistance of an identical pump was measured and found to be equivalent to about 4 feet of service line with the pump turned off.

Testing was done at five remaining residences between 9:00 and 11:00 AM on week days using the second, and final, version of the bypass manometer to measure shut-off and bypass differential heads. The velocity in the main was estimated from the shut-off head using Equation 9.2 and assuming $K_{po}=2$. The service line flows were estimated using the bypass manometer calibration curve. All flows had Reynolds numbers less than 1,000. The total equivalent length of the combination of pitorifices and fittings was calculated by assuming laminar flow using Equation 9.9. Service line lengths were either measured in the field by estimating the location of the buried main or calculated from recorded swing tie measurements. The results are summarized in Table 9.1. Discrepancies between calculated and predicted values may be due to a service line that is larger than the standard nominal ¾-inch diameter Type K copper tube or to an erroneous bypass differential head measurement. The bypass manometer could only be read to the nearest 0.5 mm WC so at low differential heads the error can be appreciable. For example, at the third site, a reading of 1.5 mm WC would yield a flow of 0.09 gpm which is 25 percent different than the flow calculated using the reported reading of 2.0 mm WC.

Table 9.1: CUC Field Study Data and Estimates

Address	Service run ft	m	Head: SO mm	BP mm	Calculated: V_m m/s	fps	Q_s gpm	L/s	L_{eq} m	ft	Predicted V_m fps
3731 Swenson	56	17.1	13.5	2.5	0.36	1.19	0.12	0.0073	3	11	1.53
3134 Amber	33	10.1	10.0	4.0	0.31	1.03	0.20	0.0126	-4	-13	0.97
1332 Prospect	32	9.8	7.0	2.0	0.26	0.86	0.12	0.0075	-1	-2	1.50
585 Sprucewood	57	17.4	50.0	16.0	0.70	2.30	0.35	0.0223	11	36	2.29
5260 Cherokee	42	12.8	17.5	4.7	0.41	1.36	0.18	0.0112	6	20	1.57

The test locations were identified on the CUC Water Flow Schematics and the predicted flows in the main were taken from the 1991 Water Network Analysis (FPE/Roen Engineers, 1991). These are shown in Table 9.1 under the heading of "Predicted". Main velocities calculated from the test data agree fairly well with those predicted by the computer model except for the third service. Either the calculated or the predicted velocity could easily be in error by 20 percent. Velocities calculated from the shut-off heads are dependent on the shape, orientation, relative size, and insertion depth of the pitorifices. Velocities predicted by the computer model are directly dependent on pipe roughness, pump performance, and water use assumptions.

Equations 4.14 and 9.9 can be combined to yield:

$$Q_s = \frac{\pi d^4 K_{po} V_m^2}{256 \nu (L_T + L_{eq})} \tag{9.10}$$

Figure 9.10 shows the predicted flows in nominal ¾-inch diameter service lines versus service run distance for different flows in the main using L_{eq}=10 m [33 ft] and T_W=5°C [41°F] for $\nu=1.5\times10^{-6}$m²/s in Equation 9.10. The data from Table 9.1 are plotted along with a line representing 0.1 gpm/100 ft run which was judged to be the limit for a 2 F° [1 C°] drop. If the acceptable water temperature drop is halved or if the heat loss rate is doubled, the line would rotate up about the origin to twice the slope. The service that is closest to being unsafe is, therefore, the first service listed in Table 9.1, with 0.21 gpm/100 ft of run and which was first served in 1984. If a service line is not buried for its entire length and is exposed to freezing temperatures along part of its distance, the heat loss rate will be significantly greater and the flow required to prevent freezing will be greater.

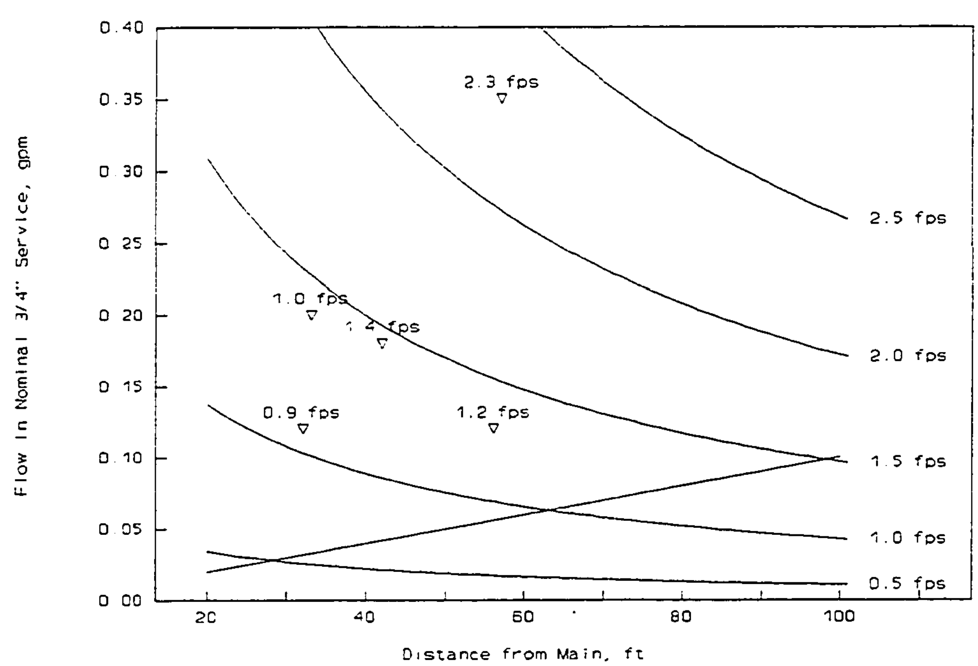


Figure 9.10: CUC Service Flow Rates Versus Service Distance and Main Flow Rate

9.5 FAIRBANKS, MUNICIPAL UTILITIES SYSTEM (MUS) SERVICE AREA

A study of services which use circulation pumps was done to provide information on plumbing and operating practices. Pitorifice shut-off heads and flows with and without pumps running were measured. This information was used to formulate plumbing recommendations, to validate pitorifice test results, and to determine whether smaller pumps could be used.

A total of nine businesses and residences in the Fairbanks MUS service area which use pumps were visited between January 12 and January 24, 1994. At each location the plumbing plan was drawn, the owners or operators were questioned, and the pressure differential across the pitorifices and flow rates with the pumps on and off were measured. Grundfos model UM15-10B7 wet rotor pumps were installed at four locations and left in place in three. Although these pumps use only 42 watts, they would have provided adequate flow in all the locations visited. Wattages of existing pumps ranged from 47 to 180 watts with an average of 113 watts. With a 42 watt pump savings would average $5/month when electricity rates are $0.10/kWh.

Three services without pumps and which froze were also visited. Pumps were installed at two of these services. Lower temperatures in the mains may be partly responsible for these freeze-ups. MUS uses waste heat from their power plant to heat water, but the power plant is being operated less as it gets older and temperatures in the main are lower than they have been in the past. Marginal services that have operated without trouble in the past may require pumps in the future.

In the late 1960s MUS installed a total of 73 pumps in homes in conjunction with water system improvements and expansion. This list of 73 locations was used to find businesses and individuals that would cooperate with the study. Selection was initially based on being able to schedule a visit during working hours. After telephoning most of the sites and visiting some, a tentative selection of 14 services was made. Data on the mains and services were then collected and testing times scheduled. Several homeowners could not be reached for scheduling and one residence was eliminated after visiting it because it would have been difficult to open the lines without property damage. A residence belonging to an MUS employee which was not on the list was visited, bringing the total number of tested locations with existing pumps to nine.

Several local thawing services were contacted and asked to report freeze-ups in service lines. Two frozen services were subsequently visited which had been identified in this manner. The third frozen service was identified by MUS.

Table 9.2 gives information on the test locations. Service line lengths were taken from planimetric drawings and service line inside dimensions (IDs) are for nominal ¾, 1, and 1¼-inch diameter Type K copper water tube. At most sites copper tube penetrates a wall or floor and brass pipe and fittings are used to complete the plumbing. At sites D and G nominal 1¼-inch diameter black iron pipe penetrated the floor and black iron pipe with black iron and brass fittings are used. At site C one pipe nipple was galvanized; it was found to be partially plugged with deposits and it was replaced with a brass nipple. Cast iron pump volutes or steel impellers were encountered at four sites and the pumps were replaced at three.

Table 9.2: MUS Field Study Data and Estimates

	Address		Service Run ft	Service Run m	Head: SO mm	Head: BP mm	Calculated: V_m m/s	Calculated: V_m fps	Calculated: Q_s gpm	Calculated: Q_s L/s	Calculated: L_{eq} m	Calculated: L_{eq} ft	Predicted V_m fps
A	710 College	¾	80	24	28	5	0.52	1.72	0.16	0.010	16	52	2.25
B	1532 10th	¾	170	52	110	10	1.04	3.41	0.21	0.013	95	313	3.61
C	1179 Gregory	¾	60	18	10	2	0.31	1.03	0.11	0.007	-1	-3	1.60
D	1915 Cushman	1¼	175	53	30	10	0.54	1.78	0.28	0.018	204	669	1.73
E	1506 Denali	¾	210	64	125	13	1.11	3.63	0.24	0.015	68	222	3.61
F	1627 Crosson	1	220	67	135	42	1.15	3.78	0.56	0.035	153	503	3.30
G	1916 Cushman	1¼	150	46	30	15	0.54	1.78	0.46	0.029	99	324	1.73
H	3199 Kiska	¾	214	65	78	8	0.88	2.87	0.19	0.012	25	83	2.39
I	1212 10th	¾	180	55	85	10	0.91	3.00	0.21	0.013	40	130	3.13
J	221 Fairwell	¾	100	30	35	25	0.59	1.92	1.04	0.066	-48	-159	1.64
K	453 Slater	¾	100	30	36	8	0.59	1.95	0.22	0.014	1	4	1.83

	Clorius Rdg: L/s	Clorius Rdg: gpm	Clorius Rdg: L_{eq} m	Clorius Rdg: L_{eq} ft	Existing Pump: Q_s L/s	Existing Pump: Q_s gpm	Existing Pump: W	Grundfos UP10-15B: Q_s L/s	Grundfos UP10-15B: Q_s gpm	Grundfos UP10-15B: L_{eq} m	Grundfos UP10-15B: L_{eq} ft	Existing Pump Data: Wet Rotor Pump?	Correct Pump Mtls?	Pump Always On?	Pumps with PO's?	Downstream tap?
A	0.011	0.17	14	46	0.172	2.73	180	0.137	2.16	2	8	N	Y	N	Y	Y
B	0.012	0.20	105	343	0.122	1.94	135	0.083	1.32	27	88	N	N	Y	N	N
C	0.003	0.04	49	159	0.148	2.35	47	0.123	1.95	26	86	Y	N	Y	N	N
D	0.005	0.08	1000	3282	0.319	5.06	83					Y	Y	Y	N	Y
E	0.014	0.23	77	252	0.117	1.85	140					N	Y	Y	Y	N
F	0.021	0.34	341	1119	0.171	2.70	85	0.103	1.63	218	714	Y	Y	-	Y	Y
G	0.022	0.34	164	538	0.286	4.53	141					N	N	Y	N	N
H	0.008	0.13	91	299	0.126	2.00	73					Y	N	Y	N	Y
I	0.013	0.20	48	156	0.127	2.02	140					N	Y	Y	Y	N
J	0.013	0.21	1	4												
K	0.008	0.12	49	159				0.115	1.82	11	36					

Main velocities were calculated using Equation 9.2 and the measured shut-off head and assuming $K_{po}=2$. Predicted main velocities were determined using the computer model and assumed values for pipe wall roughness, known pipe IDs, and known flows at pump stations. There was fair agreement between these two estimates.

The shut-off and bypass heads were used in conjunction with the bypass manometer calibration curve to arrive at an estimated flow rate for pitorifices alone. A Clorius 3VPD magnetic flow meter was used to measure flows for both pitorifices and pumps. In both cases, the service line loop was opened up and the instruments connected with two-foot or six-foot long 1-inch ID reinforced vinyl hoses. The hose and the meter losses were ignored because they had been found in past investigations to not be very significant, particularly when used in long runs. The pump was left in place during pitorifice flow measurements, because the head loss from the pump is not very significant compared to the line losses in long service runs.

The bypass manometer was often more difficult to read in the MUS system than it was in the CUC system. The manometer water levels fluctuated continuously due to unsteady flow in the main, vortex shedding of the pitorifices, the oscillation of the water in the long service lines, and changing flow rates in the main. This fluctuation was much more pronounced in the larger diameter mains and in mains with higher flow rates and could be ±25 percent or more.

The flow rates monitored with the Clorius meter were also observed to fluctuate by as much as ±25 percent about the value reported in Table 9.2, but usually they fluctuated no more than ±10 percent. This fluctuation is believed to be due to fluctuating flow rates in the mains because it was not observed with constant main flow in the full scale test section at the water plant.

Service line flows were first determined without the pumps running. There was fair agreement between the two measurement methods in most cases. The equivalent length was calculated using Equation 9.9 for both flow rates. In the case of laminar flow the Reynolds numbers were all under 1,000 so turbulent plug flows probably were not occurring. Equivalent length was also calculated for sites wherever a Grundfos UM15-10B7 pump was installed by using:

$$L_{eq} = \left[\frac{2gH_p}{V_S^2} - K_{ftgs} \right] \frac{d}{f} - L_T \qquad \textbf{(9.11)}$$

with $K_{ftgs}=5$ and the pump head, H_p, determined from a curve fit for the performance curve supplied by the manufacturer. This last determination of equivalent length is for turbulent flow and is not comparable to an equivalent length determined for laminar flow, but it does serve as a check. All the L_{eq}s can be seen to vary greatly, but with some consistency. Those calculated for site F, for example, are extremely high in all cases. One explanation is that the service line lengths were underestimated. Deposits in the lines are also a possibility, particularly when iron is present in the plumbing, but copper lines that have been in service for 38 years have been found to have no deposits. The calculation of equivalent length for the laminar case is also highly dependent on the accuracy of the measured shut-off head.

There is also poor agreement in some cases between the flows estimated with the BPM and those measured with the Clorius meter. This may be due to the fluctuating heads, fluctuations in flow rates, or inaccuracies inherent in the BPM which are particularly evident outside its calibrated range. Despite these problems, the BPM may still be a useful tool for service line evaluation by plumbers.

9.5.1 Descriptions of Sites

Many of the existing pumps had been replaced with wet rotor models which run quieter and therefore are often allowed to run continuously. Although MUS has encouraged plumbing pumps so that they work with the pitorifices, in most cases the pumps worked against them. Table 9.2 summarizes these and other plumbing issues and Figure 9.11 illustrates the actual plumbing. The following subsections provide additional information.

9.5.1.1: 710 College (Overhead Door) [A]

The existing B&G pump was found switched off. When it was switched on, the coupling was found to be broken. The coupling was replaced and the flow and wattage were measured. A new Grundfos UM15-10B7 pump was installed and the flow and wattage were measured. Electric savings were estimated to be $10/month based on $0.10/kWh. The manager has been there 13 years and said they run the pump seasonally. The only freeze-up he knew about was several years ago in January when either the coupling failed or the pump was turned off. The garage door is about 4 feet from the plumbing and cold air infiltration may have caused the freeze-up.

9.5.1.2: 1532 10th Avenue [B]

A Grundfos UM15-10B7 was temporarily installed and the flow and wattage were measured. The existing Armstrong S25 pump was plumbed downstream of the user tap and opposing the induced flow from the pitorifices. An attempt was made to reverse the volute and in the process it was discovered that the impeller was not bronze like the volute but steel. The impeller was covered with corrosion deposits, so a new B&G Series 100 pump was installed and flow and wattage were measured. The Armstrong pump used 175 watts and the B&G uses 135 watts, so the electric savings were estimated at $3/month base on $0.10/kWh. The owner said he runs the pump year round and has never had a frozen line. He believes the pump was originally installed in 1974.

9.5.1.3: 1179 Gregory [C]

The existing pump was a Grundfos model UPS15-42F which has a cast iron body and three speeds. It was set on the lowest speed and flow and power measurements were taken at that speed. A galvanized pipe nipple was replaced with a brass nipple, the existing pump with a Grundfos model UM15-10B7, and flow and power measurements were made. Both the original pump and the galvanized nipple had corrosion deposits. The owner said the pump was installed about four years ago to replace the existing B&G pump. They had turned off the B&G pump seasonally but they run the Grundfos year round.

9.5.1.4: 1915 Cushman (Tip Top Auto) [D]

The existing pump direction was found to be opposing the pitorifice induced flow. The pump was reversed which produced a small gain in pumping rate. This put the user tap upstream of the pump, but the service lines are nominal 1¼-inch diameter, so this is not significant. The pump is operated seasonally.

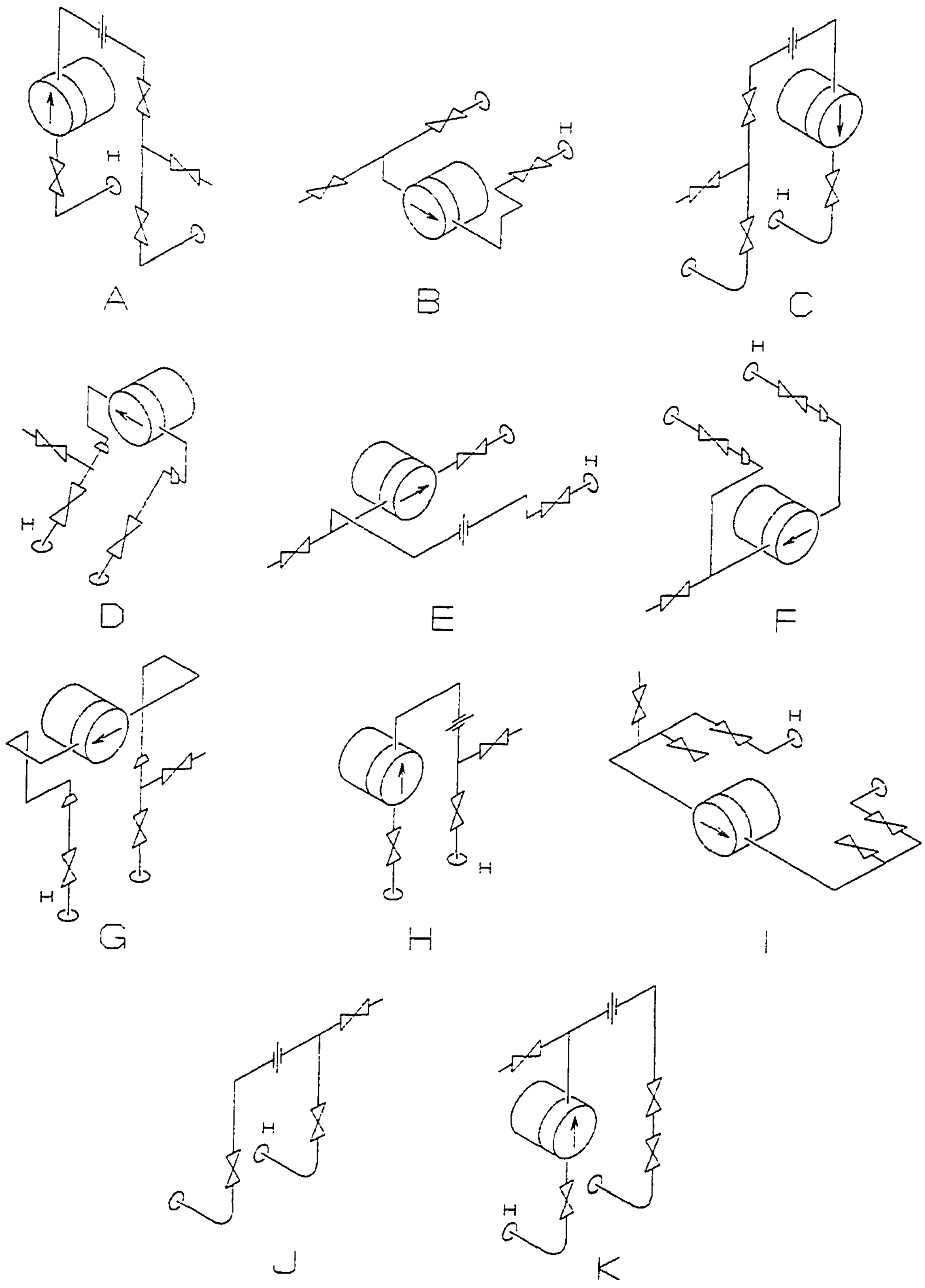


Figure 9.11: Pump Plumbing Schematics for MUS Sites Visited (H denotes leg with upstream pitorifice)

9.5.1.5: 1506 Denali [E]

The existing pump is a B&G Series 100. Flow measurements were and the power was estimated at 140 watts. The owner said the pump may be the original pump and they have not had problems except with the couplings. The pump is operated year round.

9.5.1.6: 1627 Crosson [F]

Measurements were made on the existing Grundfos UP15-42SF pump and then on a new Grundfos UM15-10B7 pump which was permanently installed. The owner said that he installed the Grundfos UP15-42SF pump several months ago when he bought the house. He replaced an existing B&G pump which was inoperative and which he believed had not worded for years.

9.5.1.7: 1916 Cushman (Tip Top Auto) [G]

The existing cast iron Armstrong S-25 pump was found switched on and warm to the touch, but not turning. The pump was plumbed so that it stuck out into the room away from the rest of the plumbing. The pump seal leaked when the pump was gently pushed. The pump was dismantled and the impeller and volute were both found to be encrusted with hard corrosion deposits. The pump was replaced with a new B&G Series 100 pump before making measurements. The plumbing was modified so the new pump would be less susceptible to damage. The pump was plumbed so that it pumped with the pitorifice induced flow and so the user tap was downstream of the pump. The Armstrong pump motor was later found to run for short periods before the thermal overload would trip. Oiling the motor bearings corrected the problem. The service manager reported that the pump is operated seasonally.

9.5.1.8: 3199 Kiska [H]

The existing pump is a cast iron body Taco 007 wet rotor type pump. The flanges were badly rusted and the pump was located where access was difficult, so the pump was not replaced.

In 1978, MUS abandoned the main in Kiska and rerouted the service to a new main in Park adding a pump to the service. The owner said the original pump was a B&G and that it failed after a few years, so he estimates they operated without a pump for 8 to 10 years. They had the Taco pump installed about four years ago, before leaving the house for a few months during the winter. They have apparently run it continuously since that time.

In March, 1994, the service line froze. After it was thawed the flows were rechecked and found to be exactly the same as those previously measured. The owners reported that they found the pump was running after they lost water flow. It was speculated that one leg of the service line may have frozen during a power outage and the other leg some days later during the night. A thermistor was installed under insulation on the service line and temperatures were logged over a two week period. Because the circulation rate in the service line is fairly high relative to the heat transfer rate, the temperatures recorded reflect those in the main fairly closely. Figure 9.12 shows that temperatures drop shortly before midnight reaching a low in the early morning (midnight is represented by vertical lines). Temperatures rise again as water use brings warmer makeup water into the loop. It is possible that with lower makeup water temperatures this pattern would be reversed with a heat loss being balanced by heat gains from loop pumping and from the service lines. This was found to be the case in the Birch Estates Loop of the College Utilities Corporation system.

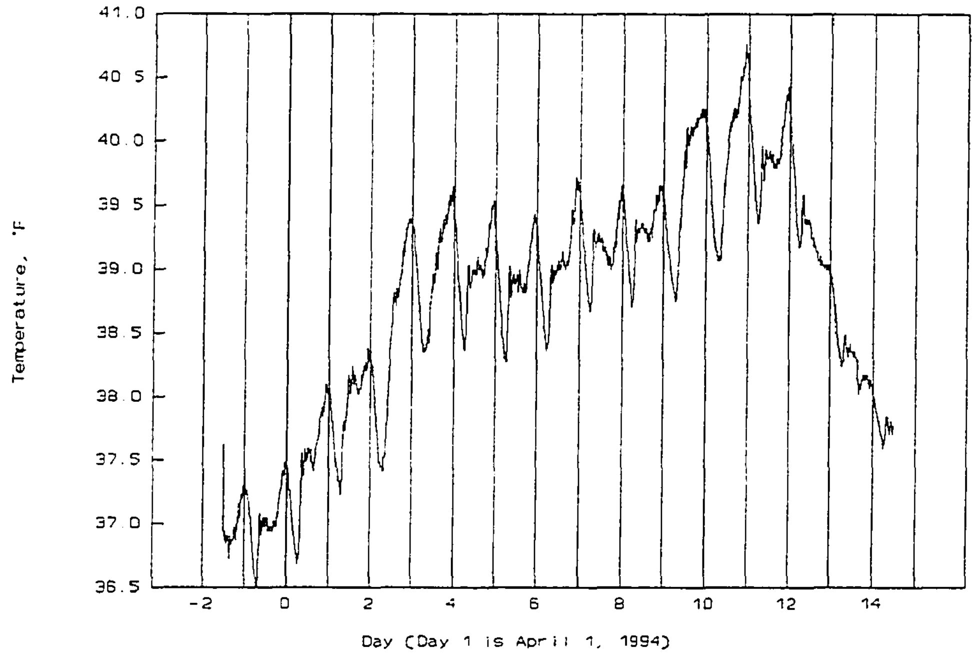


Figure 9.12: Service Line Temperatures at 3199 Kiska, Fairbanks Residence

9.5.1.9: 1212 10th Avenue [II]

Because the owner said they had problems with low pressures, the lines individually purged, taking flow measurements before and after purging. Purging was done by wasting water directly from one leg at a time into a 5-gallon pail. Over 40 gallons were removed from the upstream leg before the water cleared to a light tea color, but only 15 gallons were removed from the downstream leg before the water was clear. Apparently the sediment came from the main and settled out predominantly in the upstream leg. The flow rate was not significantly improved with purging, the real problem being the long service run. The owner said they have lived in the house 21 years. They replaced the original pump when it started making noise about ten years ago with the present B&G Series 100. He said they have had problems with low water pressure but have not had any freeze-ups.

9.5.1.10: 221 Fairwell [J]

Mr. Keith Smith of Arctic Service & Plumbing called to describe a frozen service and request a flow measurement. He said they did not have electrical continuity so they tried internal thawing with steam. They could only push the steam wand 60 feet into the 100-foot line. They then tried electrical thawing again and managed to thaw the line before continuity was again lost. Mr. Smith believed that heating the copper with the steam caused it to expand. If there was a buried coupling and it was a pack joint type and not the flare type coupling MUS requires to ensure electrical contact, the expanding copper may have made electrical contact. The flow and temperatures in the main may have been lower than normal at some time, leading to freezing in one of the legs which in turn led to the other leg eventually freezing. The owner was advised to install a pump because of the possibility that a second freeze-up would require digging up the line. Although offered a pump free of charge, the owner declined to pay for its installation, citing the long history of no freezing problems at the service.

9.5.1.11: 453 Slater [K]

Mr. Smith of Arctic Service & Plumbing called to describe a frozen service and request a flow measurement. The line froze after MUS had the main shut off for about one hour for repairs to a leak in the area. Services rarely freeze sooner than 4 hours after loss of flow, so this was unusual. The flow rate induced by the pitorifices was found to be substantially below the suggested minimum. A Grundfos UM15-10B7 pump was installed. A thermistor was attached to the line just as it entered the wall. The temperature with the pump running was 3.1°C [37.5°F], and this should have been close to the water temperature in the main. The temperature with the pump off was 1.6°C [34.9°F]. This was a 0.5 W/ft loss for one leg and presumably an overall loss of 1.0 watt per foot of run distance, suggesting colder than normal ground temperatures or poor insulation. The service line was in a crawl space and the vents were found to be open. It was possible that cold air in the crawl space removed additional heat from the service line which led to its freezing. However, the downstream leg thawed in about one hour whereas the upstream leg required over three hours, which suggests that the upstream leg froze first. This was the shallowest buried service identified. It was installed in 1984 and is 4.5 ft deep at the main and 4 ft deep for the service run. Most other services were 5 to 6 ft deep.

9.5.1.12: 2153 Bridgewater

MUS asked to have this line tested after it was reported frozen in February. No flow and no head differential were found, suggesting that either the main was valved off or the pitorifices were both gone. When a second service on the street was tested and found to also have no flow, the valves on the main were checked and a valve was discovered to have been closed. Opening the valve resulted in flow in the second service but still no flow in the first one. The owner subsequently installed a pump to ensure circulation. On April 5, 1994, MUS dug up the main. One of the service saddles was leaking and both pitorifices were found to be flattened against the side of the pipe. It was speculated that a piece of ice may have formed upstream on the main at some time in the past and that this ice was released when flow was restored to the main, causing the damage to the downstream pitorifices.

9.5.2 Summary of Findings at MUS

At sites F and H residents reported the pumps had not been operating for a number of years. Flows of 0.15 and 0.06 gpm/100 ft of run were found at these sites respectively. Although depth of burial, insulation, snow cover, water use, water temperatures in the main, and other factors can vary significantly from site to site, a value of 0.2 gpm/100 ft of run appears to be a reasonable lower limit.

MUS has a policy of having pumps installed so they pump in the direction of induced flow, but this was done in less than half of the sites investigated. Pumping direction has relatively little effect on the service line flow rate because the pump head is typically so much greater than the pitorifice head. Pumping direction also has relatively little effect on the head loss in the main as described in Section 8.5.3.

MUS has no policy on user tap location. Ideally, the tap should be downstream of the pump but this is probably not important in most cases (see the discussion on this topic in Section 8.4).

Although all the sites originally had coupled style pumps, half had wet rotor style pumps at the time of the study. The wet rotor pumps are cheaper, quieter, and require less maintenance but people also tend to leave them on year round and they are less likely to notice if they stop running because they are so quiet. Common complaints with coupled style pumps such as the B&G Series 100 were failures in the coupling and noise associated with the coupling.

Although the Grundfos UM15-10B7 had been extensively tested and compared to other models, there was one new fact brought to light by the field testing. In spite of the fact that the model UM15-10B7 was found to have approximately the same locked rotor torque as the model UP15-42SF, unlike the UP15-42SF the UM15-10B7 will stop rotating if the vent cap is removed while the pump is under system pressure. This is not recommended by the manufacturer but many people are in the habit of doing this to vent the pump and verify rotation. The proper way to vent the pump is with the pump turned off. Operation can be checked with a stethoscope, by removing the vent cap and using an ice pick to push in on the shaft so it will rotate, or by valving off the pump and then removing the vent cap.

The bypass manometer proved much more difficult to use in the MUS system than in the CUC system. Irregular flows in the mains may have been contributing to fluctuating manometer levels.

9.6 FAIRBANKS, COLLEGE UTILITIES CORPORATION (CUC) BIRCH LOOP

The temperatures in the Birch Estates Loop of the CUC system were monitored and analyzed to provide some insights into heat losses and gains in a circulating loop water distribution system. The Birch Loop was chosen for monitoring because it is the simplest and most accessible loop in the Fairbanks service area. It has no branching or shared lines and interconnections with other loops, it serves close to 70 residences but no businesses, and make-up water supply to the loop is metered. It was found that the heat loss to the ground during the winter was more than compensated by heat gains from pumping and from the exposed piping in the houses. Make-up water was actually being heated in the loop by mixing with loop water before being supplied to services. The one service line tested showed very poor circulation, presumably due to damaged pitorifices.

9.6.1 Description of the Loop and Monitoring

The main consists of 3,900 feet of nominal 6-inch diameter Class 160 uninsulated polyvinyl chloride (PVC) pipe buried about 10 feet deep. The main is located adjacent to the street on residential property easements and there are only two street crossings. The service lines are nominal ¾-inch diameter Type K copper water tube in expanded polystyrene boxes.

Pump house plumbing consists of about 30 feet of nominal 4-inch diameter schedule 40 steel pipe. A Paco 2 Hp, 1725 rpm, Type L Model 2570-7 pump with a 6.25-inch diameter impeller is used to circulate water in the loop. Make-up water is introduced upstream of the pump suction and the make-up flow rate averaged 8 gpm or about 11,000 gpd. Pressure gauges are installed on the pump suction and on the discharge volute. There is no meter for the loop circulation rate. The pump was originally selected to circulate at about 270 gpm for a flow velocity of 3.0 fps. The volume of the water in the main and services was estimated to be 6,250 gallons.

The existing loop make-up water meter was suspected of inaccuracy and was not capable of modification for automatic data logging, so it was temporarily replaced with the Clorius Model 75VPD magnetic flow meter. Custom flange adapters were machined from blind flanges to allow installation of the meter. Thermistor wells were installed in the make-up water line and the loop supply and return lines.

Flow rate was measured with a Clorius Model 3VPD magnetic flow meter, pressure differential was measured with a water/air manometer, and service line temperatures were logged using a surface mounted thermistor at one residence. Because of the very low flow rate measured, attempts were made to test flow rates at other services. Unfortunately, scheduling visits proved difficult because few residents were at home during the work day. The effort was abandoned after visiting four other residences and finding none of them suitable. The main problem was that, due to the age of the loop, many basements had been remodelled making access to piping very difficult.

The data loggers were programmed to monitor on 5-second intervals and log the average of 60 readings on 5-minute intervals. Data and calculated values were then reduced for plotting by taking averages over one-hour periods. Power use was measured with a Fluke 80i-kW current/power probe.

9.6.2 Main Loop Heat Gains and Losses

Heat is added to the loop water at the pump due to pump losses, in the loop due to frictional losses, by conduction wherever ground or air temperatures exceed water temperature, and by mixing whenever make-up water temperatures exceed loop water temperatures. Heat is lost wherever loop water temperature exceeds ground or air temperature and whenever loop water temperature exceed make-up water temperatures.

The total of pump losses and frictional heat is the power input to the pump. The power input varies slightly with the water demand but can be approximated by assuming an average demand which is uniformly distributed in the loop or by averaging the no-demand and high-demand extremes. In the case of the Birch loop, peak total demand for all services on the loop is only about 20 gpm [1.3 L/s] and is not significant. The pump curve is shown in Figure 9.13 with a calculated system curve for the no-demand case and with the manufacturer's values for pump efficiency shown.

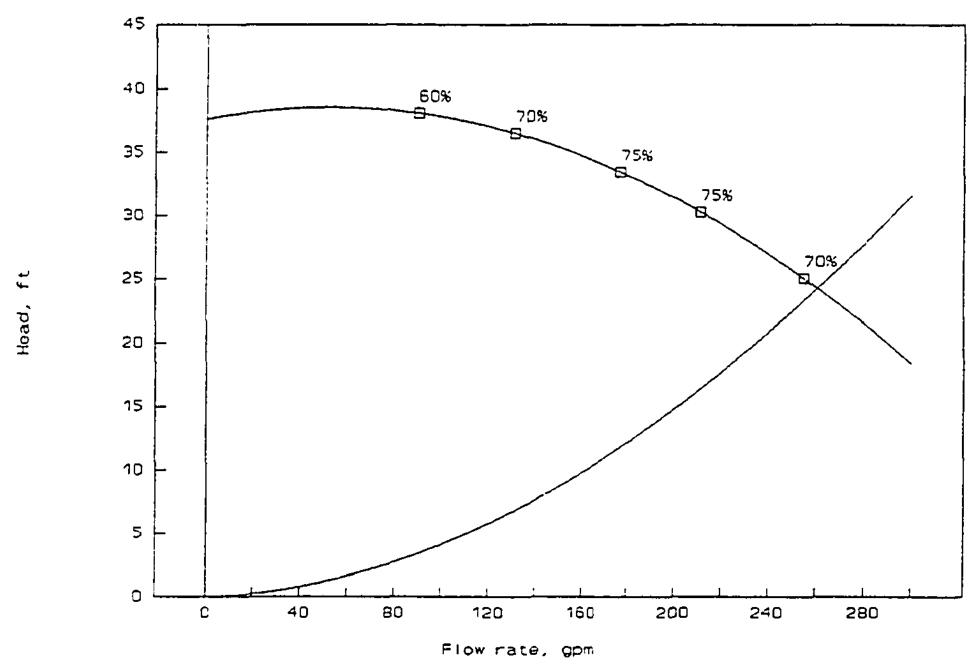


Figure 9.13: CUC Birch Loop Pump Performance and System Curves

The pump curve is a quadratic curve fit of the manufacturer's data (least-squares curve fit using orthogonal quadratic and evenly spaced points) which was extrapolated past 260 gpm. The system curve was calculated by:

$$H = \left[\frac{L_m}{D_m} f_m + N C_f + \sum K_{ftgs,m} \right] \frac{V_m^2}{2g} + \left[\frac{L_p}{D_p} f_p + \sum K_{ftgs,p} \right] \frac{V_p^2}{2g} \tag{9.12}$$

where subscripts m and p refer to main and pump house pipe, H is the head, L is the length of pipe, D is the inside diameter of pipe, f is the friction factor (calculated by Equation 4.18), N is the number of services, C_f is the loss coefficient for a pitorifice pair, K_{ftgs} are the fitting loss coefficients, and V is the average velocity in the pipe. The system curve was plotted for L_m=3,900 ft, L_p=30 ft, D_m=0.508 ft, D_p=0.336 ft, N=66 services, C_f=0.2, $\Sigma k_{ftgs,m}=4$, $\Sigma K_{ftgs,p}=2$, $\nu=1.7\times10^{-5}$ ft²/s and ϵ=0.002 ft.

Because of the absence of a flow meter on the loop, the performance of the pump could not be directly verified. Two methods of indirectly verifying pump performance were therefore employed. The suction and discharge pressures were measured at 54 psig and 70 psig when the downstream valve was closed and the pump was operating at shut-off head and at 54 psig and 62 psig when the pump was circulating water. The pressure gauges have 2-psi divisions, so assuming an accuracy of $\pm$1 psi, the shut-off and operating heads are 37 and 18 ft WC with an accuracy of $\pm$5 ft WC. While the measured shut-off head is in fairly good agreement with the pump performance curve, the operating head is low if the system curve is assumed to be accurate or the system curve is high if the pump performance curve is assumed to be accurate.

The rate of energy added to the water (the hydraulic or water power) is given by:

$$\dot{E} = \frac{Q \Delta P}{\eta_p} = \frac{QH\rho g}{\eta_p} = \frac{QH(1.9\,W\cdot gpm^{-1}\cdot ft^{-1})}{\eta_p} \tag{9.13}$$

where Q is the flow rate and η_p is the pump efficiency. Assuming the system curve is accurate and using H=18 ft WC, Q=220 gpm and η_p=0.65 the energy added is 1.2 kW. Assuming the pump performance curve is accurate and using H=18 ft, Q=300 gpm and η_p=0.60 the energy added is 1.9 kW.

The motor power use was measured at 1.5 kW. Assuming a motor efficiency of 0.85 the pump was using about 1.3 kW. This suggests that the system curve is accurate and not the pump curve. Wear on the impeller or debris lodged in the closed impeller vanes can cause degradation of pump performance. However, the pump was inspected by the CUC mechanic and found to be in good condition. For purposes of this analysis it was assumed that heat input from pumping was 1.3 kW.

A phone survey was conducted to determine whether there were any individual circulation pumps because these pumps can contribute significantly to heating water in the loop. About one forth of the residents were questioned and none reported having a pump. Because of this survey and the fact that CUC records also showed no pumps, it was assumed that none existed. Based on the previous measurement at 1466 Carr Ave., it was assumed that 6 watts/service was added for a total rate of energy into the loop from the services of 0.4 kW. Figure 9.14 shows the temperatures of make-up, supply, and return water and Figure 9.15 shows the make-up flow rates for the same period of time. The make-up temperature fluctuates widely, while the supply and return temperatures closely match one another. During the first part of the day there is very little make-up water introduced into the loop and the temperature of the water in the loop rises. Demand for water peaks in the morning and when the make-up water temperature is less than the loop water temperature, the loop water is cooled. The temperature difference between supply and return is not significant; thermistors were calibrated against one another before and after logging and found to agree within ±0.05 C°. Make-up water temperatures are lowest when make-up water flow rates are highest.

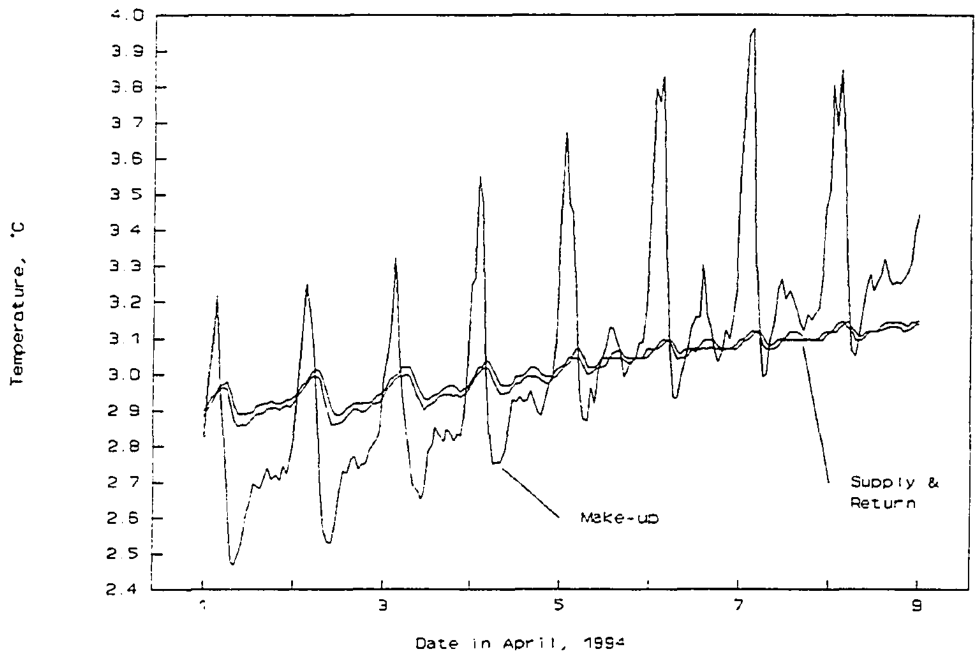


Figure 9.14: CUC Birch Loop Make-up, Supply, and Return Water Temperatures in April, 1994

The energy loss rate to the make-up water is found by:

$$\dot{E} = C_p \dot{m} \Delta T \tag{9.14}$$

where ΔT is the temperature difference between the make-up water and the loop water, C_p is the heat capacity of water, and $\dot{m}$ is the make-up water mass flow rate. The rate of energy gain to the loop water is given by:

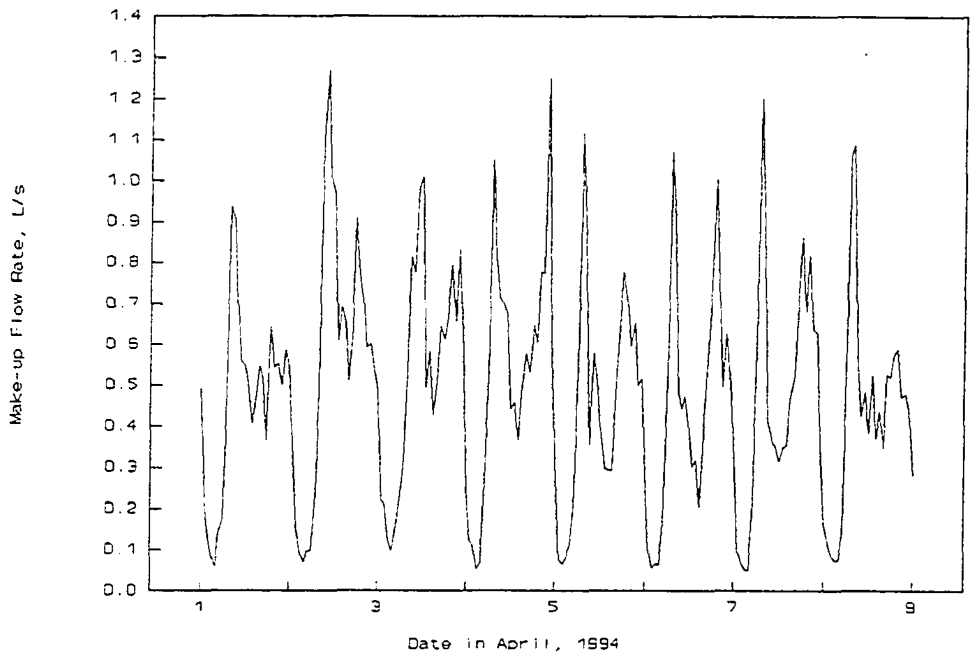


Figure 9.15: CUC Birch Loop Make-up Water Flow Rate in April, 1994

$$\dot{E} = C_p m(dT/dt) \tag{9.15}$$

where m is the mass of water in the loop and dT/dt is the rate of change of the loop water temperature.

The total energy input from pumping and heat conduction estimates was 1.7 kW. The sum of the energy loss rates to make-up water, to loop water, and to the ground must therefore equal 1.7 kW and heat flow to the ground (and pipe) can be calculated. The results are shown in Figure 9.16. There is substantial data scatter due to the very small differences in temperature used in the calculations and the measurement error of these temperatures. There is a trend towards greater heat loss to the ground shown by the line determined using a linear regression analysis of the data. This trend is expected because the average temperature of the water in the loop is increasing.

Figure 9.17 shows water temperatures for July. Similar to April, they show the loop and make-up water warming during the night, but the diurnal changes in loop temperature are much greater and the average temperature is several degrees warmer. Unfortunately, no flow rates were obtained due to problems with the analog output of the flow meter.

Figure 9.18 shows a 10-day composite of water demand flow rates for the Birch Loop for time of day. A very low flow rate is consistently seen in the early morning hours. There are peaks in the morning and evening of varying magnitude.

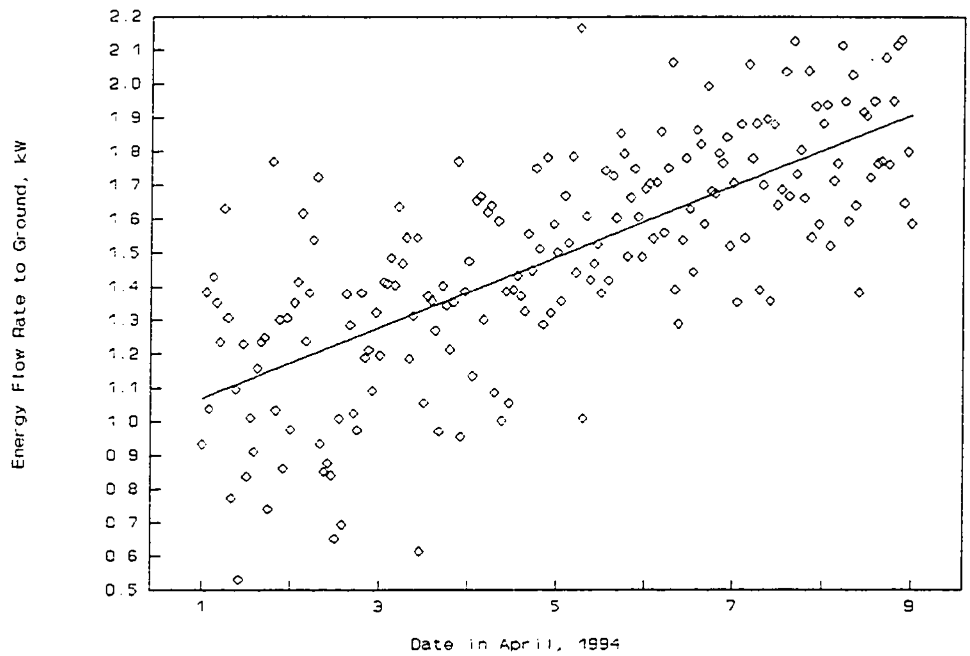


Figure 9.16: Estimates for CUC Birch Loop Heat Flow Rate to Ground in April, 1994

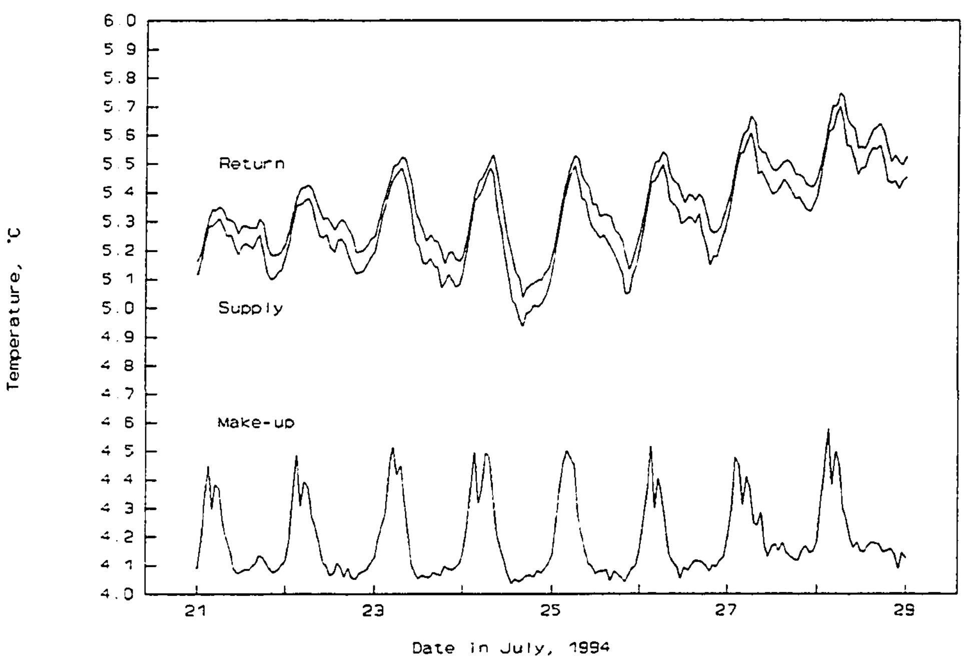


Figure 9.17: CUC Birch Loop Make-up, Supply, and Return Water Temperatures in July, 1994

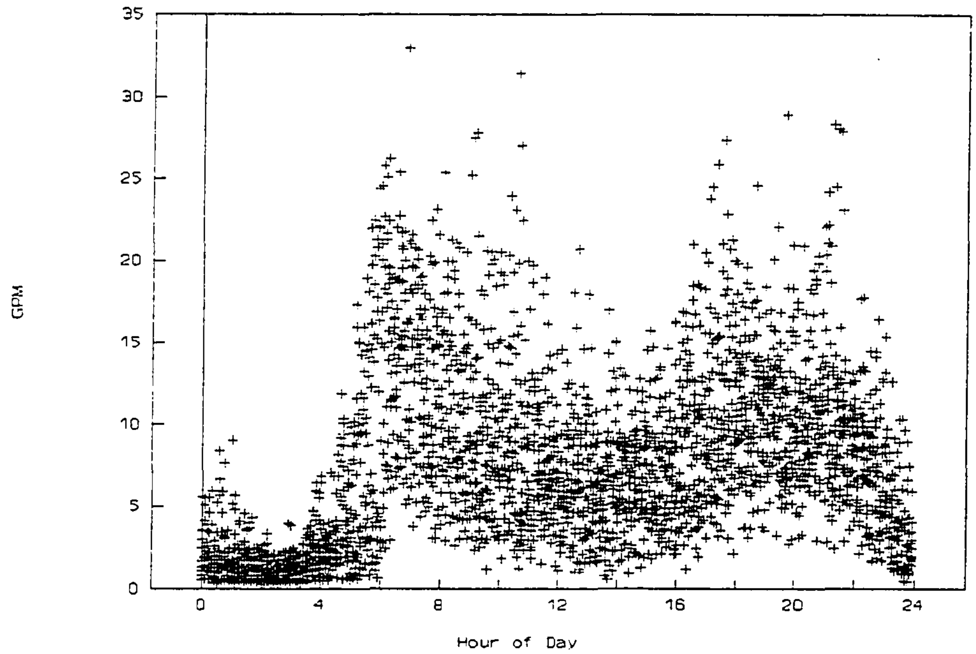


Figure 9.18: CUC Birch Loop 10-day Composite Demand, March 30 to April 8, 1994

9.6.3 Tests on Individual Services

A shut-off head differential of about ⅜-inch [10 mm] was measured at the service at Lot 64, Block B of Birch Estates. This suggests a velocity of about 1.2 fps [0.35 m/s] for a flow of about 110 gpm in the main with standard, undamaged pitorifices with $K_{po}=1.5$ using Equation 9.2. At this flow the pump efficiency is about 65 percent and head is about 38 feet so the hydraulic power to the water would be about 1.2 kW from Equation 9.13. While the power to the water is in agreement with the motor watt meter reading, the differential head is not, so it seems likely that it is the pitorifice pair that is at fault for the low differential.

A flow of 0.05 gpm [0.003 L/s] was measured at the service for a calculated average velocity, V_s, of 0.036 fps [0.011 m/s]. The sum of the total and equivalent lengths was calculated to be 210 feet [67 m] using the Darcy-Weisbach equation and assuming laminar flow. This is probably twice the actual length and is high even considering pitorifice and fitting losses which may add about 30 equivalent feet [9 m]. The calculated sum of the total and equivalent lengths is directly proportional to the differential head which may be off by 25 percent.

Figure 9.19 shows the hourly maximum and minimum temperatures recorded for a thermistor on the surface of the service line. The thermistor was installed at the inlet near the floor where the copper tube came up into the heated basement. The area around the thermistor was insulated; previous studies comparing temperatures from a surface mounted thermistor with insulation to one in a well on a service line suggest that the surface thermistor will read about 0.5 C° higher. The minimum temperatures were apparently obtained when water was withdrawn at a high flow rate from the main because they are about 0.5 C° higher than those in the main shown in Figure 9.14. The maximum temperatures were apparently obtained during times of no demand. Using Equation 9.15 and a temperature rise of 0.9 C° the energy gain rate from the main to the house is apparently 12 watts. This suggests the heat added by service lines from houses may be much greater than the 6 watts/service estimated. If the heat from service lines is greater than estimated the loss to the ground will be greater also.

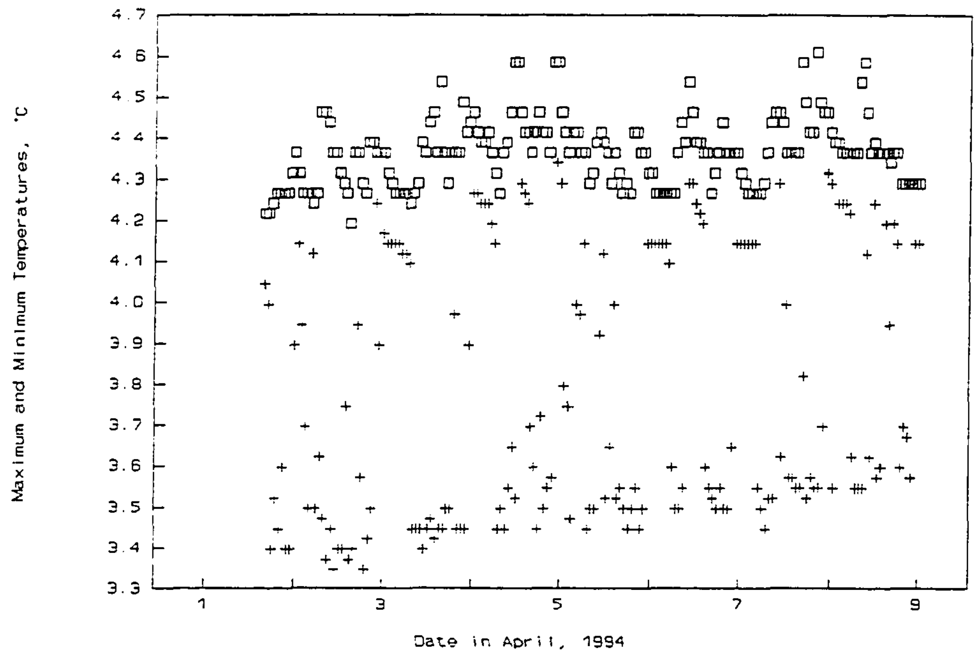


Figure 9.19: CUC Birch Loop Service Line Temperature Extremes in April, 1994
□ High; + Low

9.6.4 Summary of Findings at CUC

Heat gains from pumping and from circulation through warm environments in and near houses can more than offset heat losses to the ground.

Heat loss determinations are difficult to make because of the very small temperature differences and the effects of thermal mass and demand. While monitoring temperatures and flows in a loop can give some indication of net heat loss and gain, monitoring soil temperatures in selected locations is probably advisable for accurate heat loss measurement under specific controlled conditions.

CHAPTER 10: APPLICATION OF TEST RESULTS

Every water distribution system is unique. Climate, soils, density of services, location of service in houses, orientation of houses, lot sizes, requirements on the size of the mains, availability of waste heat, cost of construction, and fuel and electric costs can all have significant effects on the type of system that should be used and on how best to construct the system. In the following sections assumptions are made and examples of actual systems are used in calculations which illustrate design processes.

10.1 MINIMIZING STEADY STATE OPERATING COSTS

Operating costs are assumed to be limited to heating costs and electricity for pumping. Optimization is done by estimating heating and electric costs as functions of flow rate in the main, then summing these and locating the minimum total operating cost as a function of flow rate. A very simple pitorifice system operating under steady state conditions is analyzed, although any other type of arctic water distribution system could be analyzed in a similar manner. Steady state heat transfer rates are calculated for assumed soil properties under worst case and average conditions. For simplicity, services are considered to be buried and choices for only three sizes of mains, two sizes of service lines, and two insulation thickness are considered. To simplify calculations, an insulated conduit for the service lines is assumed. Although such conduits are not used in Fairbanks, they are used extensively in other places in Alaska. The system shown in Figure 10.1 is assumed to have the following parameters and choices:

Length of main, $L_M=10{,}000$ ft (neglect pump house plumbing)
Diameter of main, $D=4$, 6 or 8 inches)
Thickness of insulation on main, $t_{IM}=2$ inches
Pipe roughness factor, $C=120$
Number of services, $N_S=100$
Total length of service, $L_T=100$ ft
Equivalent length of fittings and pitorifices, $L_{eq}=30$ ft
Pitorifice performance coefficient, $K_{po}=2.4$, 2.2, or 2.0 for $D=4$, 6, or 8 inches, respectively
Internal diameter of service, $d=0.745$ or 0.995 inch
Length of service conduit, $L_{SC}=50$ ft
Diameter of conduit, $d_c=3$ inches
Thickness of insulation on conduit, $t_{IC}=3$ or 4 inches

System makeup water demand, Q_d=25 gpm or $\dot{m}_d$=12,510 lb/hr

Soil conductivity, k_s=0.5 Btu/hr · ft · F°

Insulation thermal conductivity, k_I=0.015 Btu/hr · ft · F°

Burial depth to pipe center, z=5 ft

Minimum and average ground surface temperatures, T_{GS}=−40°F and 20°F

Treated water temperature, T_a=35°F

Heat costs $10/million Btu ($0.034/kWh)

Electricity costs $0.10/kWh

Pump efficiency given by (see Appendix D): $\eta_p = 1 - 1.27\ Q_m^{-0.291}$ where Q_m=flow in gpm

Electric motor efficiency given by (see Appendix D): $\eta_m = 1 - 0.233\ (Hp)^{-0.193}$

Minimum service line return temperature, T_d=35°F

No heating of water at services, no credit for delivering warmer water to users

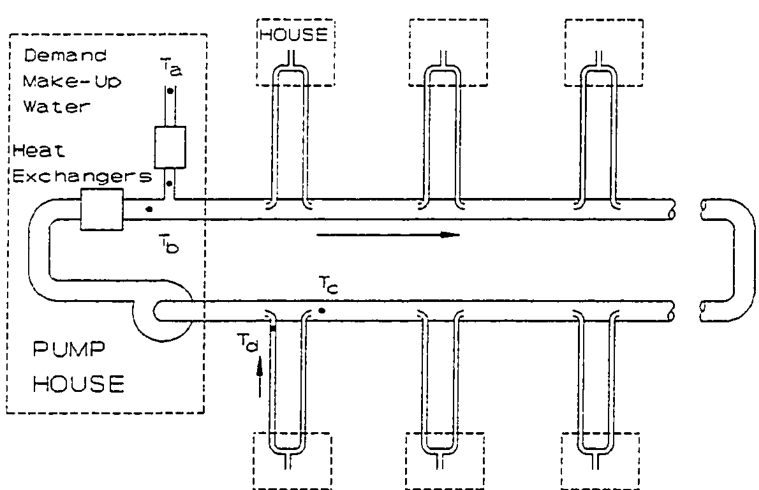


Figure 10.1: Schematic of Example Circulating System

10.1.1 Estimating Electric Costs

The electric power cost is estimated by first finding the head loss which in turn varies with the demand. Designs are usually based on maintaining a minimum return flow rate for either a maximum demand condition or an average demand condition. The average demand condition basis is used in this analysis. The Hazen-Williams formula for head loss is preferred over the Darcy-Weisbach because it simplifies the calculations. Starting with the Hazen-Williams formula in United States Customary System (USCS) units:

$$H = \frac{10.4 Q^{1.85} L}{C^{1.85} D^{4.87}} \tag{10.1}$$

where H is the head in feet of WC, Q is the flow in gpm, L length of pipe in feet, C is a constant for the type of pipe, and D is the ID in inches. (The assumption that C is constant introduces a small error: by equating the Darcy-Weisbach and Hazen-Williams formulas for head loss and solving for C for velocities between 0.5 and 1.0 fps and temperatures between 35 and 50°F, C is seen to vary up to six percent.) By assuming demand is uniform along the length, Equation 10.1 can be written as an integral:

$$\int dH = \frac{10.4}{C^{1.85} D^{4.87}} \int_0^L \left(Q_r + \frac{Q_d}{L}x\right)^{1.85} dx \qquad (10.2)$$

where Q_d is the demand flow, Q_r is the return flow, and x is the distance. Substituting y for the quantity in parenthesis:

$$\int dH = \frac{10.4}{C^{1.85} D^{4.87}} \int_{Q_r}^{Q_r+Q_d} \frac{L}{Q_d} y^{1.85} dy \qquad (10.3)$$

Finally, integrating between the limits yields an expression for head in terms of Q_r and Q_d:

$$\Delta H = \frac{10.4\,L}{C^{1.85} D^{4.87}} \left(\frac{(Q_r + Q_d)^{2.85} - Q_r^{2.85}}{2.85\,Q_d}\right) \qquad (10.4)$$

The water horsepower required by the circulation pump is next found using only the return flow rate because demand flow is added downstream of the circulation pump in the assumed system shown in Figure 10.1:

$$WHp = \frac{Q_m \, \Delta H_{Total}}{3960 gpm \cdot ft \cdot Hp^{-1}} \qquad (10.5)$$

Pump and motor efficiencies are determined using the relationships previously given, and electric power cost per hour is found by:

$$\frac{Electric\ Cost}{Hour} = \frac{(WHp)(\$/Kwh)}{1.34 Hp/Kw\,(\eta_p)(\eta_m)} \qquad (10.6)$$

The cost of heating is next determined by working backwards from the minimum service line return temperature. For the first approximation, worst case temperatures will be assumed and frictional heating contributions ignored. This will result in an upper limit for circulation rates.

First solve for $\dot{m}_s$ as a function of V_m assuming laminar flow in the service by writing Equation 9.10 as:

$$\dot{m}_s = \frac{d^2 K_{po} V_m^2}{64\, v (L_T + L_{eq})} \qquad (10.7)$$

Then solve for the service line system thermal resistance assuming negligible film and pipe wall resistances. Start with the formulas for thermal resistance of an insulated pipe and ground thermal resistance for a buried pipe (Smith, 1986):

$$R_I = \frac{\ln(r_I / r_p)}{2\pi k_I} \qquad (10.8)$$

$$R_G = \frac{\operatorname{arccosh}(z/r_I)}{2\pi k_s} \qquad (10.9)$$

Note that R_G can also be written as (Eshbach and Souders, 1975):

$$R_G = \frac{\ln\left[(z/r_I) + \sqrt{(z/r_I)^2 - 1}\right]}{2\pi k_s} \qquad (10.10)$$

Combine these to find the total system thermal resistance:

$$R_T = \frac{k_s \ln(r_I/r_p) + k_I \ln\left[(z/r_I) + \sqrt{(z/r_I)^2 - 1}\right]}{2\pi k_I k_s} \qquad (10.11)$$

Substitutions can now be made to express r_I and r_p in terms of pipe diameter and insulation thickness:

$$R_s = \frac{k_s \ln\left(\frac{d_c + 2t_{ic}}{d_c}\right) + k_i \ln\left(\frac{2z}{d_c + 2t_{ic}} + \sqrt{\left(\frac{2z}{d_c + 2t_{ic}}\right)^2 - 1}\right)}{2\pi k_i k_s} \qquad (10.12)$$

The required water temperature in the main at the last service, T_c, is then found using:

$$T_c = (T_d - T_{GS}) \exp\left(\frac{L_{sc}}{\dot{m}_s R_s}\right) + T_{GS} \qquad (10.13)$$

where the subscript GS refers to the ground surface. By assuming that the total heat loss rate from the service lines is:

$$\dot{E}_s = \left(T_c - T_d\right)\dot{m}_s N_s C_p \tag{10.14}$$

neglecting adjustments to temperature for frictional heating for simplicity because these are small (these will be considered later) and assuming no demand flow for worst case temperature drop conditions, the required supply temperature is given by:

$$T_b = \left(T_c - T_{GS} + \frac{\dot{E}_s R_m}{L_m}\right)\exp\left(\frac{L_m}{\dot{m}_m R_m}\right) + T_{GS} - \frac{\dot{E}_s R_m}{L_m} \tag{10.15}$$

where the thermal resistance of the main, R_m, is calculated similar to that for the service line.

The overall conductive heat loss rate, $\dot{E}_{bc}$, in the system can now be estimated by assuming it is equal to $(T_b - T_c)\dot{m}_m$. Heat is added to the system by the pump through frictional losses, $\dot{E}_f$:

$$\dot{E}_f = \frac{(Q)(\Delta H)}{1.556 gpm\cdot ft\cdot hr/Btu(\eta_p)} \tag{10.16}$$

The total heat that must be added to make up for conductive losses is the difference between conductive losses and the frictional heat gain. To this cost must be added the heat required to warm the demand water to the average water temperature in the system. This "Demand Heat" is given away to the consumers and can be an appreciable expense particularly when high system temperatures are maintained to compensate for very low service line circulation rates. It can be argued that this heat has value to the consumers because most of the water used in a household is either inadvertently heated by the room air or deliberately heated either through mixing (tempering) with warmer water, in a water heater, or on a stove. The first users on the loop will benefit the most because the water is warmest there. In addition, heat is added when the water service line passes through heated space and this tends to lower the heating costs. Both the value of warmer water and heating at the homes is omitted from this analysis for simplicity.

The total heat plus electric operating cost is finally computed. Figures 10.2 and 10.3 show the results for ground surface temperatures of 20°F and −40°F respectively. It should be emphasized that the −40°F case is worst case. The actual heat loss rate will be less during the unsteady state transition due to the thermal mass of the ground and steady state may never be reached. Also, the effects of moisture and snow cover are not included. Operating costs are so different between these two cases that different vertical scales are required. Figure 10.4 shows the supply temperatures required when the ground surface temperature is −40°F. Table 10.1 shows the results of varying the main and service lines sizes.

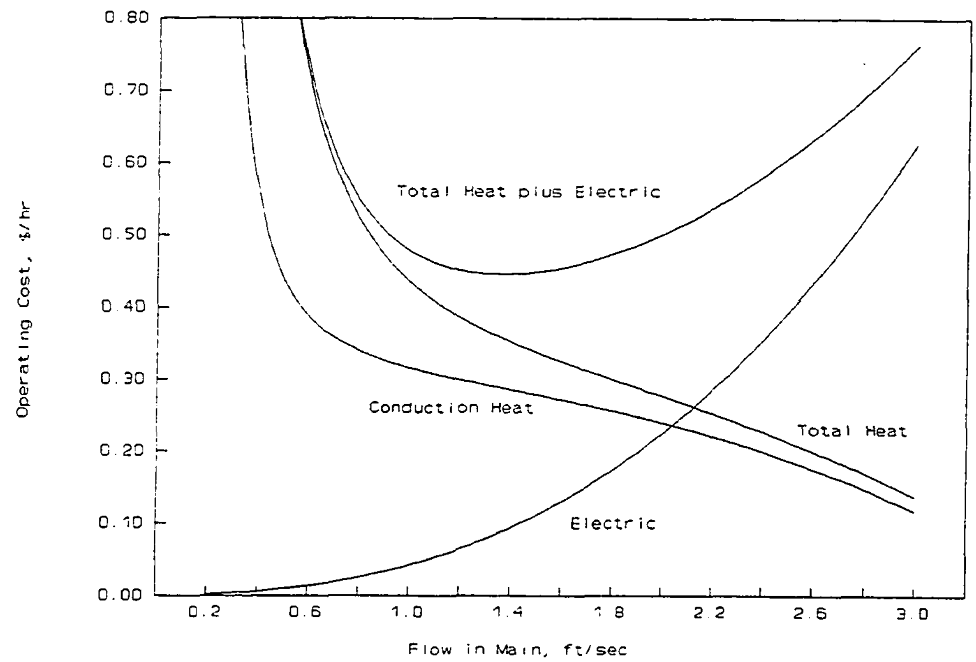


Figure 10.2: Operating Cost of 6-inch Main System with ¾-inch Pitorifices at T_{GS}=20°F

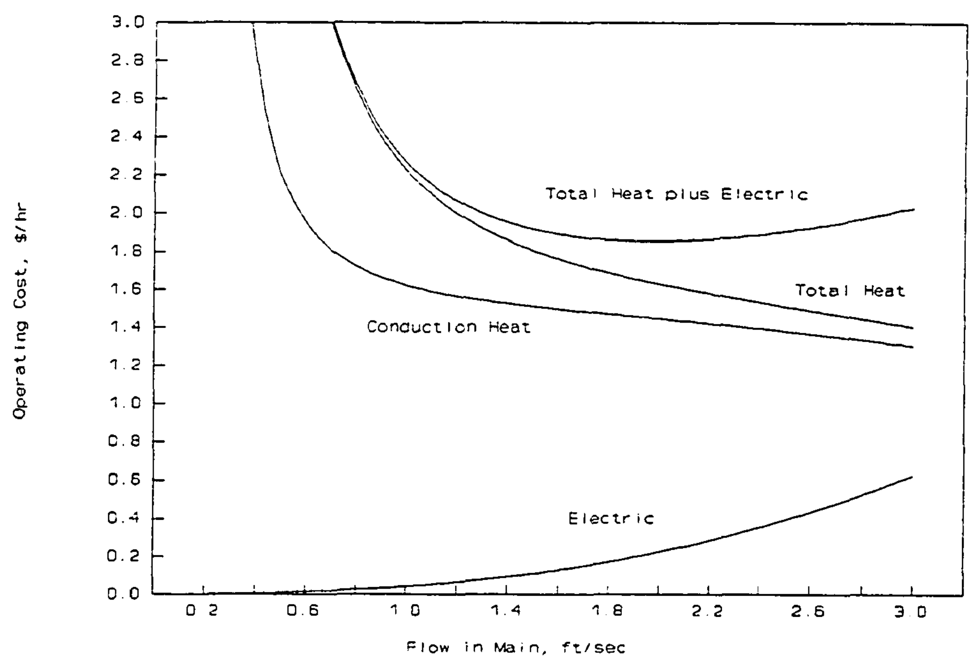


Figure 10.3: Operating Cost of 6-inch Main System with ¾-inch Pitorifices at T_{GS}=−40°F

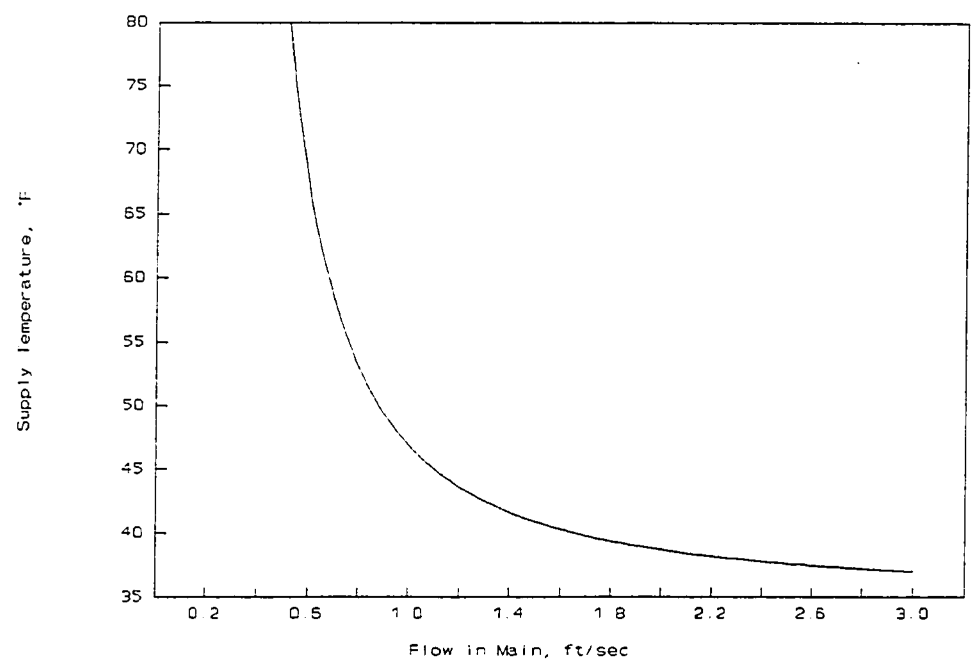


Figure 10.4: Required Water Supply Temperatures when $T_{GS}=-40°F$

Table 10.1: Minimum Operating Costs

T_{GS} °F	Size of Main	¾-inch Service				1-inch Service			
		$/hr	fps	lb/hr	T_b °F	$/hr	fps	lb/hr	T_b °F
20	4-inch	0.41	1.33	73	36.8	0.37	1.07	151	36.6
	6-inch	0.45	1.27	61	36.6	0.40	1.02	126	36.2
	8-inch	0.50	1.28	56	36.5	0.44	1.00	109	36.0
−40	4-inch	1.63	1.99	164	40.0	1.53	1.99	522	38.7
	6-inch	1.86	1.88	135	39.0	1.74	1.57	296	38.2
	8-inch	2.13	1.85	118	38.8	1.91	1.49	245	37.7

In Figure 10.2 the optimum flow rate is 1.27 fps. The heat demand is satisfied by just the energy of the pumping at some velocity in excess of 3 fps. In Figure 10.3 the optimum flow rate is 1.88 fps. Changing the demand rate will have a great effect on the optimum flow rate and if less water is delivered to consumers the minimum shifts to the left. The relative costs for heat and electricity and, less obviously, the amount of insulation will effect the optimum flow determined. The size of the main has very little influence on the optimum flow rate but a larger service line tends to result in a lower optimum flow rate as can be seen in Table 10.1.

Ideally the operator would adjust pumping and heating rates to an optimum match while monitoring the return temperatures in the critical service lines. While it may be practical and even useful to install a worst case service loop at the pump house, it is not likely to be practical or cost effective to do continuous adjustments. Single speed pumps are usually used, with a standby or booster pump which can be used during extreme cold. This means flow in the main is fairly constant, varying only as a result of changes in demand. The return temperature in the main is usually maintained at a constant minimum value by activating a heat exchanger on the return line. A conservative approach would be to require the return flow to equal or exceed the worst case return flow and the return temperature (approximately T_c) to equal or exceed the corresponding worst case return flow temperature. Operating this way will not result in minimum costs but will minimize control problems. The seasonal cost can then be estimated by using the average ground surface temperature. Assuming an average ground surface temperature of 20°F results in average operating costs of \$0.45/hr and \$0.40/hr for the ¾-inch and 1-inch diameter services with 6-inch diameter main respectively. An annual savings of \$2.20/service using 1-inch instead of ¾-inch diameter services may not be enough to warrant its additional installed cost of about \$70. With fewer services per 1,000 feet of main, higher relative electric costs (lower relative heating costs), heat added at services, or a higher minimum required temperature in the service lines, 1-inch diameter services would be favored. Also, the 1-inch diameter services will have about a 50 percent longer time to freeze in the event of a loss of circulation, all other conditions being equal.

If there are only a few long services, it may be advantageous to use even larger diameters for only the longest services. Larger diameter service lines may be preferable to heat tracing and individual circulation pumps. Less capital investment and lower operating and maintenance costs may be obtained.

Electric costs may not always increase with the size of the main for a given flow velocity in the main at the last service. Larger pipes accommodate return flow and additional demand flow with less effect on head loss. Larger capacity pumps are usually more efficient, and the cost for power delivered to the water is proportional to $D^{0.83}$ as shown by the following derivation:

$$\$/hr \propto Q\,\Delta H \qquad \textit{for constant efficiency}$$

$$\$/hr \propto \frac{Q\,Q^{1.85}}{C^{1.85}D^{4.87}} = \frac{\left(VD^2\right)^{2.85}}{D^{4.87}} \tag{10.17}$$

$$\$/hr \propto D^{0.83} \qquad \textit{for constant } V \textit{ and } C$$

10.1.2 Optimizing Insulation

Increasing the insulation on the mains and particularly on the service lines will result in lower operating costs. Not only is less heat lost by conduction, but the required supply temperature and service line flow rates can be lower. This in turn results in lower pumping costs and also permits the use of a lower required water supply temperature. However, the benefits of increased insulation must be balanced against the additional capital cost of the insulation minus any capital savings from using smaller pumps, boilers, and mains.

Determining the optimum insulation thickness is an iterative process. An insulation thickness for mains and services is assumed, then the required minimum flow and an operating cost is estimated. An incremental change in insulation is then made and the new operating cost estimated with an adjustment to reflect the change in the cost of the insulation.

An additional inch of insulation on the service lines would cost an extra $65/service (based on Fairbanks contractor prices of $4.20 and $5.50/LF for 3-inch and 4-inch thick cover) and result in an annual savings of less than $1/service. This is a return of less than two percent on the investment. The extra insulation will increase the time to freeze, however, in the event of a loss of circulation.

Once a circulation rate is determined, the pumping costs will be constant throughout the operating season. Heating costs, however, will vary over the heating season. The average, and not the worst case heat loss should be used for these estimates. Of course, the equipment associated with heating will still have to be sized for the worst case.

10.1.3 Design Considerations

Generally the design engineer will want to put the high water users first on the loop and the low elevation services last. The heat loss for the worst case must be calculated for recently backfilled trenches and not for the in-situ soils, but a less conservative value should be used for long term operating cost estimates. Warmer water and larger service and main diameters result in a slower cooling rate of the water in the event of a loss of circulation. Two-speed pumps or a standby booster pump may be an economical compromise between a single high rate or variable rate pump. It is prudent to consider sizing the pump and motor so that a larger impeller may be accommodated if needed due to future system expansions. It is good practice to balance pump house plumbing losses against the cost of larger pipe and fittings. The use of orifice or venturi flow meters on the main should be considered as these are relatively robust, simple, and trouble free compared to propeller or turbine meters.

10.2 CASE STUDY OF BRANCHED MAIN SYSTEM WITHOUT HEATING

Fairbanks uses 6, 8, 10-inch and larger diameter mains for water distribution to allow fire flow requirements to be met. In Fairbanks, the water distribution mains are not simple loops but are actually networks with sub-networks. The minimum flow velocity in any main with pitorifice services is designed to be at least 1.8 fps and some mains will have flows significantly greater than that depending on how well balanced the network is.

One of the simplest parts of the MUS system in Fairbanks is the Executive Park sub-network served by Pump Station No. 14. To further simplify analysis, the effects of make-up water, flows to and from the three interties with other parts of the Fairbanks system, and head loss contributions by fittings are ignored. The network is shown schematically in Figure 10.5 with the dimensions given in both Tables 10.3 and 10.4.

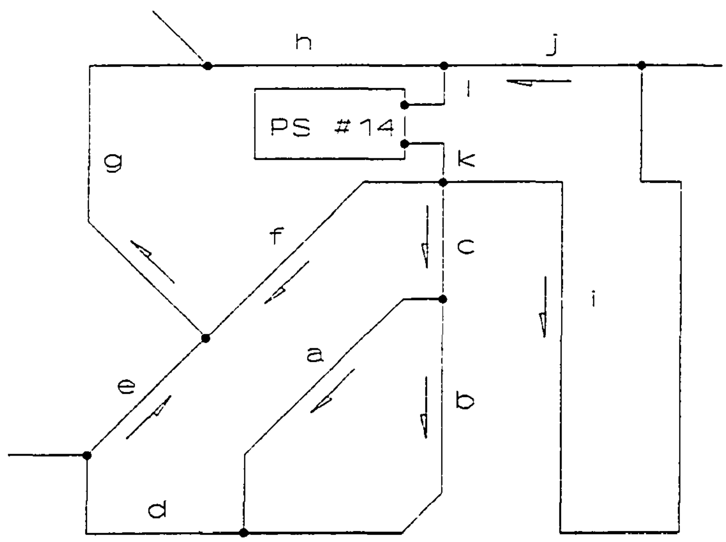


Figure 10.5: Executive Park Flow Schematic

The critical leg of the network is "a". Assuming C=140 and stipulating a 1.8 fps flow velocity in "a" results in a head loss of 2.9 ft. This same head will appear across "b" resulting in a flow velocity of 2.2 fps. The combined flow from "a" and "b" must flow through "c", "d", and "e" resulting in a flow velocity of 3.2 fps in these lines. The total head across lines "c", "b", "d", and "e" is 10.5 ft. This same head must appear across "f" resulting in a flow velocity of 4.4 fps in that leg. The combined flows from "e" and "f" go through "g" and "h" so head losses can be computed for these. The total head loss through "f", "g", and "h" must equal the total loss through "i" and "j". The flow through "i" and "j" is found by:

$$Q = C\left[\frac{\Delta H}{10.6}\left(\frac{(D_1 D_2)^{4.87}}{L_1 D_2^{4.87} + L_2 D_1^{4.87}}\right)\right]^{0.54} gpm \tag{10.18}$$

The total flow through Pump Station No. 14 is then found by summing flows from "h" and "j". Finally, the head losses in "k" and "l" are calculated. The results of this cascading series of conditions are summarized in Tables 10.3 and 10.4 as the "No Balancing" cases for minimum flows of 1.8 and

Table 10.2: Executive Park Sub-network Flows for 1.8 fps Minimum Case

C = 140
T of water at PS #14 = 50 F
T of ground surface = -50 F
Soil conductivity = 2 Btu/hr·ft·F°
Service line heat loss = 309 Btu/hr

No Balancing

Leg	D (in)	L (ft)	PO pairs	Q gpm	dH (ft)	V fps	Btu/hr gain	Btu/ hr·ft	Btu/hr loss	dT	Temp	Leg
a	6.22	1380	23	170	2.9	1.8	314	13	25677	-0.3	49.7	a
b	8.33	1315	26	377	2.9	2.2	696	17	30296	-0.2	49.8	b
c	8.33	310	4	548	1.3	3.2	474	17	6483	-0.0	50.0	c
d	8.33	1050	17	548	4.6	3.2	1607	17	23027	-0.1	49.7	d
e	8.33	385	4	548	1.7	3.2	589	17	7752	-0.0	49.7	e
f	6.22	960	15	417	10.5	4.4	2803	13	17553	-0.1	49.9	f
g	8.33	1365	0	965	16.9	5.7	10488	17	23096	-0.0	49.8	g
h	13.31	610	0	965	0.8	2.2	478	25	15209	-0.0	49.7	h
i&j				273	28.1							i&j
i	5.50	3095	64	273	28.1	3.7	4933	12	57714	-0.4	49.6	i
j	13.31	480	0	273	0.1	0.6	10	25	11968	-0.1	49.5	j
k	8.33	40	0	1239	0.8	7.3	626	17	677	-0.0	50.0	k
l	8.33	160	0	1239	3.1	7.3	2503	17	2707	-0.0	49.7	l
		Totals	153	1239	32.0		25522		222158			

WHp = 10.0   Total heat gain = 30379   Btu gain/loss = 0.11

η_p = 0.84

η_m = 0.86 $/yr = $6,116

With Balancing (extra resistance added to legs "b", "f", and "i")

Leg	D (in)	L (ft)	PO pairs	Q gpm	dH (ft)	V fps	Btu/hr gain	Btu/ hr·ft	Btu/hr loss	dT	Temp	Leg
a	6.22	1380	23	170	2.9	1.8	314	13	25677	-0.3	49.7	a
b	**8.33**	**1315**	**26**	**306**	**2.9**	**1.8**	**564**	**17**	**30296**	**-0.2**	**49.8**	**b**
c	8.33	310	4	476	1.0	2.8	318	17	6483	-0.0	50.0	c
d	8.33	1050	17	476	3.5	2.8	1077	17	23027	-0.1	49.6	d
e	8.33	385	4	476	1.3	2.8	395	17	7752	-0.0	49.6	e
f	**6.22**	**960**	**15**	**170**	**8.7**	**1.8**	**955**	**13**	**17553**	**-0.2**	**49.8**	**f**
g	8.33	1365	0	647	8.1	3.8	3350	17	23096	-0.1	49.6	g
h	13.31	610	0	647	0.4	1.5	153	25	15209	-0.0	49.6	h
i	**5.50**	**3095**	**64**	**133**	**17.1**	**1.8**	**1467**	**12**	**57714**	**-0.8**	**49.2**	**i**
j	13.31	480	0	133	0.0	0.3	1	25	11968	-0.2	49.0	j
k	8.33	40	0	780	0.3	4.6	167	17	677	-0.0	50.0	k
l	8.33	160	0	780	1.3	4.6	670	17	2707	-0.0	49.5	l
		Totals	153	780	18.8		9431		222158			

WHp = 3.7   Total heat gain = 11542   Btu gain/loss = 0.04

η_p = 0.82

η_m = 0.83 $/yr = $2,410

Services are assumed to be equivalent to a 3" pipe w/ 3" insulation
All other pipe assumed to have 2" insulation
Insulation is assumed to have a k = 0.011 Btu/hr·ft·F°
Bury depth is assumed to be 5 feet to pipe center
Service line lengths assumed to be 50 feet one way

Table 10.3: Executive Park Sub-network Flows for 0.9 fps Minimum Case

C = 140
T of water at PS #14 = 50 F
T of ground surface = -50 F
Soil conductivity = 2 Btu/hr•ft•F
Service line heat loss = 309 Btu/hr

No Balancing

Leg	D (in)	L (ft)	PO pairs	Q gpm	dH (ft)	V fps	Btu/hr gain	Btu/ hr•ft	Btu/hr loss	dT	Temp	Leg
a	6.22	1380	23	85	0.8	0.9	44	13	25677	-0.6	49.4	a
b	8.33	1315	26	189	0.8	1.1	97	17	30296	-0.3	49.6	b
c	8.33	310	4	274	0.4	1.6	66	17	6483	-0.0	50.0	c
d	8.33	1050	17	274	1.3	1.6	223	17	23027	-0.2	49.4	d
e	8.33	385	4	274	0.5	1.6	82	17	7752	-0.1	49.3	e
f	6.22	960	15	209	2.9	2.2	389	13	17553	-0.2	49.8	f
g	8.33	1365	0	483	4.7	2.8	1455	17	23096	-0.1	49.5	g
h	13.31	610	0	483	0.2	1.1	66	25	15209	-0.1	49.4	h
i&j				137	7.8							i&j
i	5.50	3095	64	137	7.8	1.8	684	12	57714	-0.8	49.2	i
j	13.31	480	0	137	0.0	0.3	1	25	11968	-0.2	49.0	j
k	8.33	40	0	619	0.2	3.6	87	17	677	-0.0	50.0	k
l	8.33	160	0	619	0.9	3.6	347	17	2707	-0.0	49.3	l
		Totals	153	619	8.9		3540		222158			

WHp = 1.4   Total heat gain = 4401   Btu gain/loss = 0.02
η_p = 0.80
η_m = 0.79   \$/yr = \$959

With Balancing (extra resistance added to legs "b", "f", and "i")

Leg	D (in)	L (ft)	PO pairs	Q gpm	dH (ft)	V fps	Btu/hr gain	Btu/ hr•ft	Btu/hr loss	dT	Temp	Leg
a	6.22	1380	23	85	0.8	0.9	44	13	25677	-0.6	49.3	a
b	**8.33**	**1315**	**26**	**153**	**0.8**	**0.9**	**78**	**17**	**30296**	**-0.4**	**49.5**	**b**
c	8.33	310	4	238	0.3	1.4	44	17	6483	-0.1	49.9	c
d	8.33	1050	17	238	1.0	1.4	149	17	23027	-0.2	49.3	d
e	8.33	385	4	238	0.4	1.4	55	17	7752	-0.1	49.2	e
f	**6.22**	**960**	**15**	**85**	**2.4**	**0.9**	**132**	**13**	**17553**	**-0.4**	**49.6**	**f**
g	8.33	1365	0	323	2.2	1.9	465	17	23096	-0.1	49.2	g
h	13.31	610	0	323	0.1	0.7	21	25	15209	-0.1	49.1	h
i	**5.50**	**3095**	**64**	**67**	**4.8**	**0.9**	**204**	**12**	**57714**	**-1.7**	**48.3**	**i**
j	13.31	480	0	67	0.0	0.2	0	25	11968	-0.4	47.9	j
k	8.33	40	0	390	0.1	2.3	23	17	677	-0.0	50.0	k
l	8.33	160	0	390	0.4	2.3	93	17	2707	-0.0	48.9	l
		Totals	153	390	5.2		1308		222158			

WHp = 0.5   Total heat gain = 1685   Btu gain/loss = 0.01
η_p = 0.78
η_m = 0.75   \$/yr = \$389

Services are assumed to be equivalent to a 3" pipe w/ 3" insulation
All other pipe assumed to have 2" insulation
Insulation is assumed to have a k = 0.011 Btu/hr·ft·F°
Bury depth is assumed to be 5 feet to pipe center
Service line lengths assumed to be 50 feet one way

0.9 fps respectively. If pump efficiencies of 0.84 and 0.80 and motor efficiencies of 0.86 and 0.79 for large and small pump systems respectively are assumed, the seasonal electric costs are \$6,120 and \$960 for minimum flows of 1.8 and 0.9 fps respectively for a savings of \$5,160 in electric costs.

Unlike village pump houses, most pumping stations in Fairbanks do not have boilers for heating the water. Heat is added to the water by frictional losses from pumping and by the make-up water. The frictional heat gain may be significant in some cases. If smaller pumps are used, additional water may have to be circulated from the main water plant where water could be heated with waste heat from the coal fired power plant. This makes it difficult to determine a cost for making up the heat that is lost when large pumps are replaced with smaller ones. However, it is reasonable to assume the upper limit would be the price the city charges for hot water district heating.

The difference in frictional heat gain in this example is 26,900 Btu/hr. Assuming heat at \$7/million Btu (the city's charge for district hot water heat) the additional seasonal cost of heating water would be \$810. Assuming all the lost motor energy is needed for building heat and any decrease in this must be made up by electric resistance heating, the additional cost of heating the building would be \$650. This leaves a net savings of \$3,700/year or \$24/year/household.

Assuming 1-inch diameter service lines were used instead of ¾-inch, an average distance of 50 feet, and a savings of \$1,200 for smaller pumps, an additional one time investment of \$70-\$1,200/153=\$62 per household would be required.

10.2.1 Heating Considerations

A significant difference between systems in Fairbanks and the villages is that almost all the remote pump stations in Fairbanks are electrically heated. These electric heaters are seldom needed because the pump motors provide heat. As long as this situation exists, and too much heat is not supplied, there is little justification to use high efficiency motors in these pump stations. High efficiency pumps are desirable, however, because the water is heated much more cheaply with waste heat from the power plant than with electric energy by way of pumping.

Maximum heat loss can be estimated by modeling service lines as 50 foot lengths of 3-inch diameter pipe with three inches of insulation. The following are also assumed: ground surface temperature of $-50°F$, water temperature of $50°F$, five foot burial to all pipe centers, two inches of insulation on distribution mains, and soil and insulation thermal conductivities of 2 and 0.011 $Btu/hr \cdot ft \cdot F°$, respectively. Using these conditions and the following formula for heat loss rate per foot for buried, insulated pipe derived from Equation 10.11, a total of 222,200 Btu/hr was calculated:

$$Q = \frac{2\pi k_I k_S \Delta T}{k_S \ln(r_I/r_P) + k_I \ln\left[(z/r_I) + \sqrt{(z/r_I)^2 - 1}\right]} \qquad \textbf{(10.19)}$$

The frictional heating of water due to pumping can be cause water temperatures to rise when demand and heat losses are low. However, the design must be for worst case demand and heat loss, and in such cases the frictional heat gain will often be less than 20 percent of the heating requirement.

Assuming four people at each of the 153 services and 100 gpd/capita demand, there would be 21,000 lb/hr of water entering the pipe. Pumping supplies an estimated 30,400 Btu/hr for the 1.8 fps case so the make-up water must supply 191,800 Btu/hr. This results in a 9.1 F° temperature drop for the make-up water (bringing it down to 50°F).

If pumping supplied only 4,400 Btu/hr, as it would in the 0.9 fps case, then the make-up water must supply 217,800 Btu/hr resulting in a 10.4 F° temperature drop for the make-up water. If the make-up water is not supplied warmer than when the larger pumps were used, the system will equilibrate to about 49°F instead of 50°F.

The Fairbanks water system sub-networks are designed to allow circulation through them back to the main water treatment plant where heat could be added. Maintaining the temperature in the sub-networks could therefore be independent of water use within each sub-network. To be truly meaningful, any analysis must take into consideration the whole system and not just an isolated portion of it. The possibility of adding heat exchangers at the main plant must also be considered.

10.2.2 Requiring 1-inch Diameter Service Lines

Consumers will need to pay about $70 extra for 1-inch versus ¾-inch diameter service lines. However, they will then have a service that will contain more water and therefore take longer to freeze if circulation were stopped, and a service that should have more water circulating through it so it will be less likely to freeze. Water rates will also be lower because the expense of operating the system would be less.

10.2.3 Balancing the Flow in the Mains

When a loop branches and one leg has less resistance than the other, flows in that leg will be higher than the minimum required to induce adequate flow in the service lines. The analysis of the Executive Park subdivision showed that to be the case in three places. If flow restriction is added to these mains, the overall flow and head loss can be reduced resulting in energy savings. This has been done in a few places, but it was done without measuring flows and by just relying on theory and assumptions. If flows in the main are calculated from the shut-off head of pitorifices or if flows in services are directly measured then correct balancing of the mains can be done. Table 10.3 shows potential savings for the 1.8 fps minimum flow case of $3,700/yr for per household savings of $26/yr.

Balancing can be done by throttling a valve, or by bypassing a normally closed valve with a line containing an orifice plate in it. The later method is preferred because a partially open valve is often inadvertently fully opened or fully closed by the operator. An orifice plate can also supply a direct means of measuring the flow in the main. With balancing, it may be possible to replace the pump motor with a slower running, lower horsepower model and keep the original pump.

If both the existing pump and motor are kept after balancing is done, the overall flow in the mains will be greater, offering a greater margin of safety. However, the effect of adding extra heat to the water is probably more significant than the effect of the extra velocity because velocity increases only with the cube root of the energy input. Rather than heat the water by pumping, it would be better to use immersion electric heaters, because they could be thermostatically controlled and there would be no waste heat from the motors. The motor heat serves to heat the pump station building but there is a lack of control so over-heating may result.

10.3 CHOOSING BETWEEN PITORIFICES AND DISTRIBUTED PUMPING

A lot of factors can influence the choice between pitorifices and small pumps for service line circulation. In general, pumps are preferred when the service runs are long, when there are relatively few services, where mains are large and used for transmission and velocities can be low, or when electric energy is inexpensive. Pitorifices are useful where waste heat is available or where power supplies are unreliable (although battery backup for pumps is an option).

10.3.1 Operating Costs

The total electric energy requirement will usually be higher with individual circulating pumps but the heating requirements will be less because much of the electric energy ultimately goes to heating water. Electric energy usually costs around three times as much as heat energy to produce, and waste heat is often available for less than one-tenth the cost of electricity. In some systems, either no heat is added to maintain service line temperatures or waste heat is used. In such cases pitorifice systems realize a much greater savings than distributed pumping systems.

When making a more precise comparison between cost for electricity and heating, the commercial versus residential rates and demand and service charges for electricity need to be considered as well as the efficiency and life cycle cost of a boiler. Subsidies should not be applied to energy costs because a subsidy is only a means of having someone else pay the bill; the cost still exists. The majority of the electric energy cost for a distributed pumping system is paid by the consumer as part of their electric bill. While the total electric cost may be greater than that for a pitorifice system, the utility may pay far less. Again, the cost still exists so it should be considered when making engineering decisions and advising clients.

The annual operating cost of a distributed pumping system can be significantly reduced by having homeowners turn off their pumps during the summer months. A service may freeze if the homeowner turns the pump off too soon or forgets to turn the pump back on in the Fall. Deposits may form while the pump is off which could prevent restarting the pump and lead to damage to the motor windings if the rotor is not manually rotated to free it. Finally, a wet rotor pump may have water condense on windings when cold water is in the pump and the motor is not running. This water can lead to motor failure as well.

Balancing of flows is not always feasible in a pitorifice system if there is branching. Some branches may operate at much higher than necessary flow rates unless a head loss device is added. Adding a head loss device will result in lower operating costs, but these will not be as low as when no balancing is required. While significant, the increase in operating cost for even an unbalanced system still does not usually result in pitorifices costing more than individual circulating pumps to operate.

High volume water users should be put at the beginning of a loop so that demand water is not pumped a great distance. This is not always possible, and the impact of demand may be greater on a pitorifice system than on a distributed pumping system because a pitorifice system uses higher flow rates in the mains.

Longer service runs require higher flow rates in the mains in a pitorifice system when all other parameters are the same. If there are only a few long service runs these should be equipped with individual pumps in addition to the pitorifices to allow the use of lower flow rates in the mains.

10.3.2 Capital Costs

By assuming a common base cost for saddles, corporations stops, service line, and fittings, an incremental cost above the base cost can be used for comparison purposes. At a minimum, a distributed pumping system needs a small pump which will cost about $100 uninstalled. Such a system can also economically use ¾-inch diameter service line.

The incremental cost of using two pitorifices instead of two corporation stops was estimated to be $10. The incremental cost of 1-inch diameter copper water tube service line over a ¾-inch line was estimated to be $0.44/ft and the incremental cost for larger saddles, pitorifices, and fittings was estimated to be $50. Finally, an incremental cost per service for the increased size of the main pump for pitorifice systems may be assumed. For very long mains, intermediate pump stations may be needed in a pitorifice systems to keep pressures from being too great, adding to their cost.

If service line diameter is kept at ¾-inch, a pitorifice system is cheaper to install than a distributed pumping system. If service line diameter is increased to 1-inch for just the pitorifice system, the first costs will be comparable, but comparison should be made on the basis of the present worth of future replacements. In the case of the service lines, there will be no replacement for the life of the water system. However, the small pumps will need to be replaced about every 10 years. Also, the larger service lines will better accommodate a water system pressurized fire protection sprinkler system. Sprinkler systems are becoming more popular for residential use and those pressurized from the water system are relatively inexpensive and very reliable.

10.3.3 Reliability

Failures result in additional costs. If water pressure is lost and the distribution system drains, individual circulating pumps may run dry, overheat and seize and water in service lines in either type system may freeze. Services in a pitorifice system are much more susceptible to freezing than those in a distributed pumping system if circulation in the main is lost while water pressure is maintained. This is not usually a problem because the pump house usually has alarming and backup pumps or power.

Using the same minimum service line temperature for both systems results in the same time-to-freeze in the event of a loss of circulation, but there is a great deal of difference in freeze susceptibility in the event of loss of insulation. Because service flow rates are so much higher in a distributed pumping system, water in the service line is far less likely to freeze if insulation is damaged or cold air infiltration occurs. A distributed pumping system may also be designed to operate at a lower temperature, reducing heating costs at the expense of the time-to-freeze allowance.

Both systems can fail due to power failures. All motors may suffer damage due to poor power conditions. In a pitorifice system the pump motors can be more economically protected and backup power supply provided to ensure circulation is maintained during a power failure. Individual circulating pumps fail individually and the service line may freeze if an operating automatic back-up is not in place, but individual circulating pumps will ensure flow in a service even if flow in the main stops. If the community power supply is very dependable, individual circulating pumps will function well. In such a case it may also be useful to have a flow alarm switch and even have the switch automatically activate a heat trace in the event of pump failure although this will add to the installed cost. Where there are frequent power outages, a pitorifice system may be more successful because backup diesel powered pumps or backup electric generation at the water plant will ensure circulation and no frozen services.

Larger service line diameters result in longer times to freeze in the event of circulation stopping. A 1-inch diameter line will take more than 50 percent longer for the same temperature drop as a ¾-inch line due to the greater mass of water and pipe wall, assuming the same heat loss rate. If larger lines are used it may be advisable to require all lines to be the same large size to reduce inventory requirements.

10.3.4 Complexity

A pitorifice system is mechanically simpler than a distributed pumping system; it has fewer moving parts that need monitoring, maintenance, and replacement. Both systems have installation concerns. Wet rotor individual circulating pumps must be oriented to ensure no air is trapped in the pump and the pump may need to be bled or the rotor may need to be manually rotated before start-up. Pitorifices must also be correctly oriented during construction to ensure they face up and down stream. Pitorifices are commonly made with the opening on the same side as the corporation stop handle so the correct positioning can be verified before backfilling, but mistakes in construction or installation have been made.

The design of a new pitorifice system or the expansion of an old system can be more demanding than that of a distributed pumping system, especially if there is a lot of branching or if optimizing for electric and heat costs is attempted. An expansion may require an intermediate booster station to maintain minimum flows in a main without excessive line pressures. This may be a particular concern where houses would not otherwise need pressure reducing valves at the service. If the water system is also providing fire protection, pressure reducing valves at the homes will often be required anyway (¾-inch pressure reducing valves cost about $34). Also, while operating at higher pressures will increase problems with leaks, it will be possible to save on the capital cost of mains sized for fire flow.

If a community is already familiar with a certain type of system, changing it or adding another type may result in problems because of contractor/operator/homeowner unfamiliarity and increased inventory requirements. For example, while some of the mains in a distributed pumping system may operate at flow rates which will permit the use of pitorifices, it may be more practical to not use them.

10.3.5 Corrosion, Wear, and Damage

A pitorifice is made from copper water tube . A very high velocity of water flowing past the pitorifice may increase corrosion rates through erosion corrosion. It should be emphasized that corrosion of any kind has generally not been a problem with pitorifices. None of the pitorifices that were installed in Fairbanks, Alaska in 1953 and subsequently recovered when the wood stave main was replaced in 1991 showed any sign of wear or corrosion loss.

Individual circulating pumps are not directly susceptible to corrosion from the water because types used in potable water (open) systems have stainless steel or bronze construction, but corrosion byproducts from other sources may plug the impeller or jam the rotor. This is not usually a problem, particularly when pumps are operated continuously.

Individual circulating pumps are subject to corrosion from their environment. If they are exposed to humid conditions while not operating, water vapor may condense on the electrical parts due to cooling by water. Damage has occurred to pitorifices when ice formed in mains has broken loose or when construction debris has been carried into the pitorifices. Pitorifices have been found with rags wrapped around them.

10.3.6 Future Expansion or Modification

A distributed pumping system is easier to expand. Expanding a pitorifice system can be a challenge, however the option to go to individual circulating pumps is always available. A pitorifice system can be converted to a distributed pumping system whereas it is impractical to convert a distributed pumping system to pitorifices.

10.3.7 Liability

In the event of a loss of power in the case of individual circulating pumps, residents may feel the electric company is liable for a service line freezing, and in the event of a loss of circulation in the main in the case of pitorifices they may feel the water utility is liable. In the event of a power failure, residents should be aware of the fact and open their faucets slightly to avoid freezing. Loss of circulation in a main will usually be quickly detected by the utility and backup pumps or generators started before there is any danger of freezing. In both systems, loss of a water supply or water pressure can lead to freezing in services. When there is a possibility of freezing, residents should be informed so they can run faucets, turn on heat traces, or drain lines.

10.4 POSSIBLE FUTURE DEVELOPMENTS

There is considerable potential for improvement of small circulating pumps. Smaller or more energy efficient pumps are certainly possible; smaller pumps, even if less efficient, may more closely match the requirements for service line circulation and therefore use less energy. Varying the flow rate or the temperature of water in the loop in response to the weather and time of season can result in considerable savings for a pitorifice system. Copp et al. (1956) reported that flows in the dual pipe system in Flin Flon, Manitoba were varied in response to weather conditions. They installed remote sensors and telemetry in the 1980s to allow varying the heat rate as required. Pumping rates in communities with pitorifices are occasionally increased during particularly cold times. As a rule, however, while the pumping rate and return temperatures may be varied at the operators discretion, they are not varied automatically. The use of a service loop at the pump house may allow better control of temperatures and better use of pumps. With a monitoring loop, the water can be heated to maintain the temperature in the service line at a certain value. Microprocessor controllers might also be programmed to adjust temperatures in response to time of year or temperature sensors for air or ground temperatures.

Monitoring the flow and temperature in worst case services and sending this information over a residential phone line to the pump house is also a possibility. Measuring low flow rates without introducing a significant pressure drop can be done with ultrasonic, magnetic, or thermal flow meters and temperatures can be monitored with thermistors. Data loggers are available which can be remotely accessed or which will automatically dial out and transmit information whenever certain preprogrammed conditions are met. Instrumenting a few key service locations may be a cost effective way to monitor the system as a whole.

Temperature controlled internal heat tracing or heat tracing which is integrated with the pipe may someday allow lines to be operated without circulation. Many locations in Greenland use electric heat tracing for active freeze protection. Use of heat tracing instead of circulation has also been proposed for intermittent fill systems for arctic regions (James and Robinson, 1979; James, 1980b).

CHAPTER 11: CONCLUSIONS

It was hypothesized at the beginning of the study that a more complete understanding of the functioning of pitorifices would lead to more economical designs and operation for cold region water systems. It was shown that the head loss formally assumed to be due to pitorifices in a main is about ten times too high. It was also shown that the best performance can be obtained from one of the simpler shapes to construct. A better performing pitorifice was not developed as a result of the study; it was demonstrated that better performance can be achieved by using different shapes for the upstream and downstream pitorifices, but installing two differently shaped pitorifice was not judged practical for construction. Field investigations showed unexpected high heat gains at residences. These findings, in combination with the test data collected and the theoretical framework that was developed, allow better designs.

It was also hypothesized that small pumps might be competitive with pitorifices. It was found that most pumps used for service line circulation are oversized and could be replaced by smaller pumps for substantial savings. It is conceivable that pumps will be developed in the future that will make a distributed pumping system more economical to own and operate than a pitorifice system, but this study confirms that pitorifices still offer the best alternative in most situations today.

The first objective of this study was to develop techniques to measure pitorifice performance in the field. As a result of this study an inexpensive device for measuring flow in a service line, the bypass manometer described in Chapter 5, was developed and shown to be effective. In the community of Lower Kalskag, service line flow measurements using this equipment alerted the operator to problems with flows in the mains that were easily remedied to avoid service line freeze-ups.

A much more accurate method of measuring flows from both pumps and pitorifices was used for the first time as a result of this study. A low head loss magnetic flow meter, also described in Chapter 5, was proven useful in measuring flow rates and trouble shooting service line and main line problems in Fairbanks.

The second and third objectives of this study were to characterize performance of existing pitorifice shapes and to seek a more efficient shape. It was found that there was little difference in performance for most shapes. The use of an incomplete bend and cut (IB&C) shape is suggested because it is cheaper to make and offers a slight advantage in performance over other shapes. However, the traditional bend, cut and turn (BC&T) and incomplete bend, cut and turn (IBC&T) shapes may still find use in all HDPE pipe systems. A prototype all HDPE pipe/pitorifice service tee was developed during the course of the study and is described in Appendix C.

While it is intuitively obvious that the head developed across a pair of pitorifices should increase with size and insertion depth, this study quantified these effects as well as the resulting head loss in the main (see Chapter 7). In fact, in a departure from previous studies, shut-off head was determined for the first time. This, combined with estimates for losses in the pitorifice when water is flowing, allows engineers to apply the data more generally than had been previously possible.

Guidelines based on extensive testing and theory now allow the engineer to depart with confidence from the "rule of thumb" which dictated 2 fps velocities in mains and a 60-foot limit to service runs and make significant reductions in the energy requirements for pumping. Design engineers can determine the service line mass flow rate needed for the expected heat loss and the allowed return temperature and then calculate the required flow rate in the main which provides that service line flow. Investments in 1-inch or even larger diameter service lines will be paid back in a few years in energy savings in many cases.

The fourth objective was to research the competing technology of small pumps used in a distributed pumping system. Historically, pumps manufactured for hydronic heating systems have been used for this purpose with little thought given to the special requirements of service line water circulation. As a result of this study several types of pumps never used before on service lines were investigated and installed in services for long term evaluation. Also, the amount of heat from the pumps which goes to the water being circulated was determined for representative types of pumps for the first time. The design engineer now has a wider selection of pumps to choose from and a better understanding of the selection criteria.

Pitorifices will in most cases be the economical and practical choice. However, small pumps will remain practical in very small systems, in retrofits and expansions of existing systems, and for supplementing pitorifices in longer service runs. The smaller, more energy efficient pumps identified in this study should be considered for these applications. Operating costs can be cut fifty percent or more with these pumps with no appreciable loss in performance.

The fifth and final objective of this study was to present the information in a way that would be useful to the design engineer. Appendix G is a summary of those aspects of this thesis which are of most concern to the design engineer and can be used as a stand alone reference. More detail is presented in Chapters 7, 8, and 9 which describe testing of pitorifices and pumps, and in Chapter 10 which deals with example analyses of water distribution systems based on representative assumptions and practical concerns in designing a system. In addition to this, Appendices C through F discuss other aspects of system design which should be of use to the design engineer.

It is not the author's intent to dictate exactly how to design a water distribution system for a cold region. There are too many variables and approaches to consider, and economics is not the only criteria for making choices. It is not practical to put a dollar value on every aspect of a circulating water system. It is even difficult to arrive at reasonable estimates of operating and capital costs for comparison purposes. The final decision should be based on a good understanding of all the factors that may be significant. It is the author's hope that this thesis will further the engineering community's understanding of some of these factors.

APPENDIX A: DATA FROM PREVIOUS STUDIES

Table A.1 gives the parameters of the previous studies, Figure A.1 shows the induced service line mass flows calculated from data from these studies. Using mass flows instead of velocities provides a common basis of comparison. Figure A.2 shows the shapes used in the 1953 study, and Tables A.2, A.3, and A.4 show data from the 1953 and 1977 studies (Johnson, 1978; Harris, 1987; Page, 1953; Spehalski, 1986; Westfall, 1953). See also Sections 3.1, 3.4 and 3.5 and Figures 3.1 through 3.4.

Table A.1: Summary of Previous Pitorifice Studies

	1953	1977	1987 & 1988
Main			
Length, feet	60	20	10
Nominal size	4 (a)	6	4
Actual ID	(4.0)	(6.065)	3.68
Material	Wood stave	(Sch 40 Steel)	HDPE
Deter. flow by	Weigh tank	V-notch weir	Turbine meter
Deter. dH by	Ker. mano.	(Water mano.)	Water mano.
Flows, fps	1.1 to 4.4	2.02 and 4.2	0.99 to 2.25
Service			
Length, feet	40+ (b)	25	100 & 200
Nominal size	¾	¾, 1, 1¼	1
Actual ID	0.79 (c)	(d)	1.076
Material	Copper (e)	Copper (f)	HDPE
Deter. flow by	Head loss	Timed dye	Timed electrolyte
Deter. dH by	Ker. mano.	(Water mano.)	Water mano.
Pitorifice			
Shape	BC&T	(BC&T)	IBC&T
Nominal size	¾	¾, 1, 1¼	1
Thread type	Wood stave	(AWWA) (g)	NPT
Actual OD	0.75 (h)	(0.75, 1.125, 1.375)	0.9375
Insertion	To center	To center	1.5 past center
Kv	0.25	0.10 to 0.18	0.37
Separation	12	8	18
Orientation	90° apart	Same plane	Same plane
Temperature	58°F	(60°F)	(60°F)
No. of Data Pts.	12 (i)	7	28
Other	(j)(k)		

Notes:
(a) All dimensions are in inches unless otherwise noted. Items in parens are assumed values.
(b) Two 20-foot lengths of copper tube were connected with hose to the pitorifices.
(c) ID also reported as 0.782 inch and 0.066 feet [0.792 inch].
(d) Not given.
(e) Refereed to as "¾-inch thin walled copper pipe" so it is probably Type K copper water tube with ID=0.795 inch and not pipe with ID=0.920 inch.
(f) Could be Type K, L, or M copper water tube.
(g) These were supplied by Fairbanks which uses AWWA thread
(h) Dimensioned this way in plans and corresponds to OD for nominal ⅝-inch tube, however shown connected with hoses to nominal ¾-inch copper water tube so may be same OD=0.875 inch.
(i) Three points for insertion to 0.5 inch before center were also taken, as well as data on other shapes.
(j) Head discharge tests were done on 39-foot length. Assume pressure taps on service were 6 inches from pitorifices.
(k) A specific gravity of 0.8 was given for the kerosene .

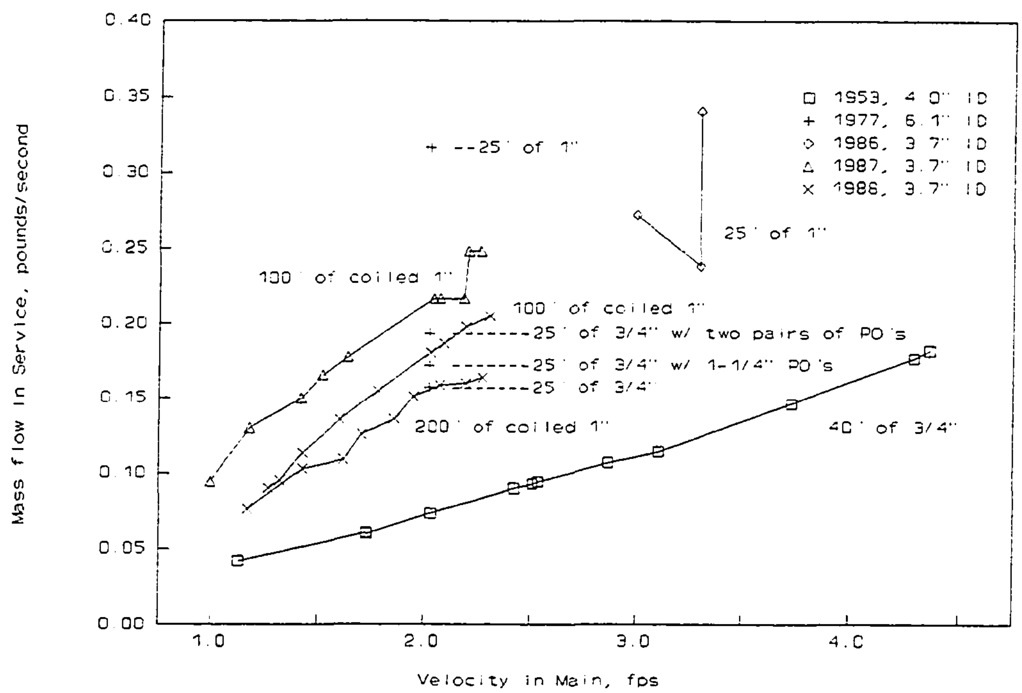


Figure A.1: Data From Previous Pitorifice Studies

Table A.2: Data from 1977 Pitorifice Study (Johnson, 1978)

V_m fps	H_m ft WC	tube size	PO size	H_s ft WC	V_s fps	Re_s
2.02	0.242		none			
2.02	0.268		none			
4.20	0.820		none			
2.02	0.282	1¼	1¼	0.109	0.972	8286
4.20	0.979	1¼	1¼	0.481	2.162	18431
2.02	0.314	1¼	1¼	0.110	0.972	8286
2.02	0.270	1	1	0.107	0.885	6030
2.02	0.254	¾	¾	0.098	0.748	3816
2.02	0.285	¾	1¼	0.151	0.921	4698
2.02	0.288	¾	2x¾*	0.124	0.819	4178

*Two upstream and two downstream facing pitorifices

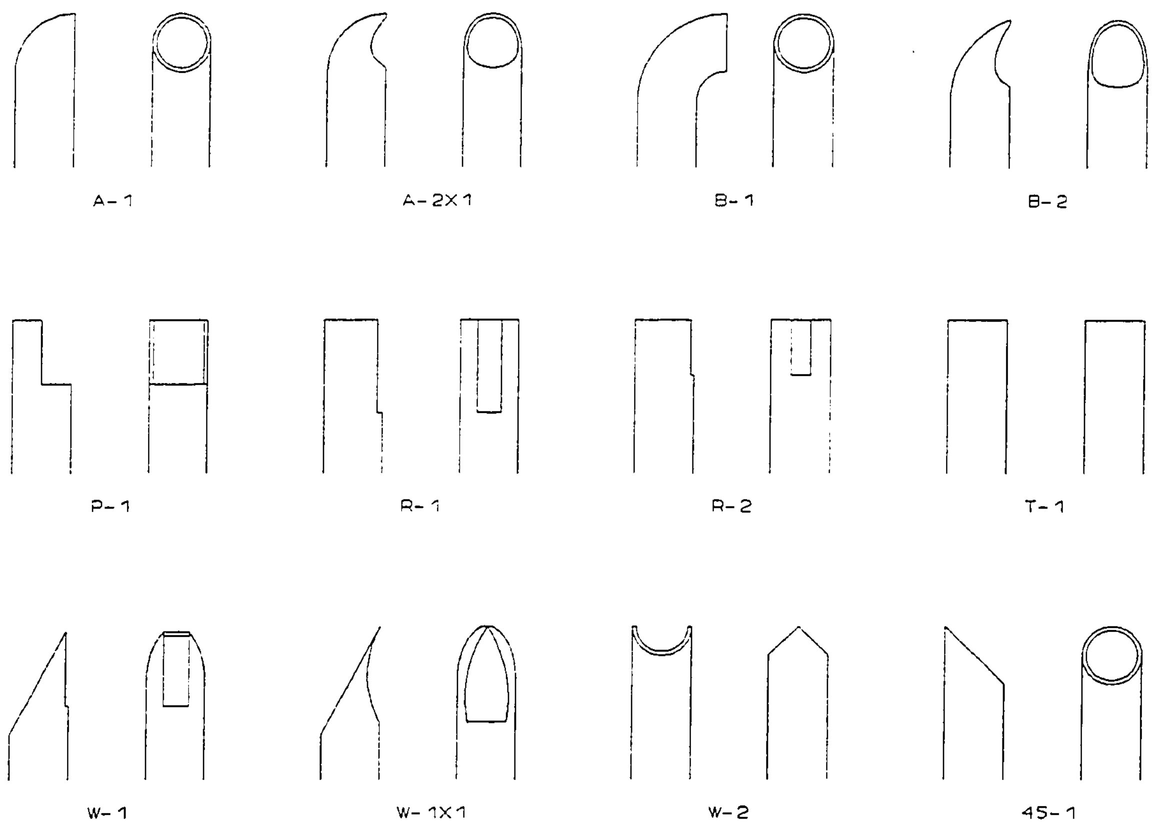


Figure A.2: Shapes Used in 1953 Study (Page, 1953; Westfall, 1953)

Page and Westfall did not discuss the manufacture of the various pitorifice shapes and their drawings lacked detail and dimensioning. Figure A.2 was prepared from their drawings. Shapes A-2×1, B-2, and W-1×1 are presumed to be shapes A-1, B-1, and W-1 after turning in a lathe. Shapes P-1, R-1, R-2, and T-1 are presumed to be open at the top. Shapes W-1 and W-1×1 were always shown with the same profile but the face-on view varied. Because the angle was shown so sharp and the opening for W-1 was always shown squared, these are presumed to be made by cutting a tube at a 28° angle and soldering on a plate.

Table A.3: Data From 1953 Pitorifice Study (Page, 1953; Westfall, 1953)

	Pitorifice			(a) Man.				V_m	(b) Man.	V_s	Re
	Up	Dn	Run	in.	lbs	sec	gpm	fps	in.	fps	
	W-1	W-2	1	1.70			175.0	4.45	25.7	0.872	4670
			2		2000	112.0	128.5	3.28	14.5	0.591	3164
			3		2000	135.2	106.4	2.72	10.1	0.481	2572
			4		2000	190.0	75.7	1.93	4.9	0.322	1726
			5		1000	146.6	49.1	1.25	2.0	0.213	1139
			6		1000	201.2	35.8	0.91	1.1	0.179	957
	R-1	W-2	1	1.70			175.0	4.45	21.4	0.764	4092
			2		2000	111.4	129.2	3.30	12.1	0.531	2841
			3		2000	159.0	90.5	2.31	5.8	0.356	1908
	P-1	W-2	1	1.70			175.0	4.45	23.1	0.807	4320
			2		2000	101.4	141.9	3.62	16.2	0.634	3393
			3		2000	155.0	92.8	2.37	7.1	0.405	2169
			4		2000	244.5	58.8	1.50	2.8	0.243	1301
	A-1	W-2	1	1.70			175.0	4.45	26.8	0.900	4818
			2		2000	126.5	113.7	2.90	12.2	0.533	2855
			3		1000	106.0	67.9	1.73	4.1	0.292	1564
	B-1	W-2	1	1.70			177.0	4.52	27.1	0.908	4858
			2		2000	109.2	131.8	3.36	16.8	0.649	3473
			3		2200	167.4	94.5	2.41	8.8	0.448	2398
			4		1000	136.0	52.9	1.35	2.6	0.236	1261
(c)	B-3	W-2	1	1.70			175.0	4.45	28.1	0.933	4992
			2		2000	125.0	115.1	2.94	13.5	0.566	3030
			3		1000	118.2	60.9	1.55	3.5	0.270	1443
	R-2	W-2	1	1.70			175.0	4.45	25.8	0.875	4683
			2		2000	115.7	124.4	3.18	14.6	0.594	3177
			3		1000	124.1	58.0	1.48	3.2	0.258	1382
	W-1x1	W-2	1	1.70			175.0	4.45	25.3	0.862	4616
			2		2000	160.1	89.9	2.30	7.4	0.413	2209
	A-2x1	W-2	1	1.70			175.0	4.45	26.2	0.885	4737
			2		2200	132.2	119.7	3.06	13.3	0.561	3003
			3		1000	86.6	83.1	2.12	6.7	0.390	2090
	B-2	W-2	1	1.70			175.0	4.45	25.2	0.860	4603
			2		1500	112.6	95.8	2.45	8.4	0.438	2344
			3		1000	107.6	66.9	1.71	4.1	0.292	1564
	B-2	A-2x1	1	1.70			175.0	4.45	24.8	0.850	4549
			2		1000	68.2	105.5	2.69	9.6	0.468	2505
	B-2	T-1	1	1.70			175.0	4.45	25.0	0.855	4576
			2		2000	148.2	97.1	2.48	8.3	0.435	2330

Table A.3: Data From 1953 Pitorifice Study (Continued)

	Pitorifice Up	Dn	Run	(a) Man. in.	lbs	sec	gpm	V_m fps	(b) Man. in.	V_s fps	Re
(d)	B-2	A-2x1	1	1.70	2000	84.2	170.9	4.36	26.6	0.895	4791
			2		1500	102.6	105.2	2.69	11.0	0.503	2693
(e)	W-1x1	B-2	1		2000	84.8	169.7	4.33	28.7	0.948	5073
			2		1000	75.4	95.4	2.44	10.2	0.483	2586
(f)	W-1x1	R-2	1		2000	84.4	170.5	4.35	29.0	0.955	5113
			2		1000	85.1	84.5	2.16	7.7	0.420	2250
(g)	W-1x1	R-1	1		2000	84.0	171.3	4.37	30.0	0.981	5248
	W-1x1	R-1	1		2000	83.8	171.7	4.38	24.4	0.840	4495
(h)	B-2	B-2	1	1.70	2000	85.6	168.1	4.29	24.8	0.850	4549
			2		4000	168.4	170.9	4.36	25.8	0.875	4683
			3		2000	118.2	121.7	3.11	12.8	0.548	2935
			4		1500	159.2	67.8	1.73	4.1	0.292	1564
			5		1000	162.6	44.2	1.13	1.7	0.202	1079
			6		1500	108.5	99.5	2.54	9.0	0.453	2424
			7		1000	106.0	67.9	1.73	4.0	0.288	1544
			8		1000	90.5	79.5	2.03	5.7	0.353	1887
			9		1500	109.6	98.5	2.51	8.8	0.447	2391
			10		2000	98.5	146.1	3.73	18.9	0.702	3756
			11		1500	113.4	95.2	2.43	8.2	0.433	2317
			12		1500	96.0	112.4	2.87	11.4	0.513	2747
(i)	B-2	B-2	1		2000	126.6	113.7	2.90	10.0	0.478	2559
			2		2000	84.5	170.3	4.35	23.1	0.807	4320
			3		1000	120.4	59.8	1.53	2.6	0.236	1261
	45-1	45-1	1		2000	84.0	171.3	4.37	20.2	0.734	3930
			2		2000	120.5	119.4	3.05	10.3	0.486	2599
			3		1500	130.5	82.7	2.11	4.9	0.322	1726
			4		1000	147.7	48.7	1.24	1.6	0.198	1058

NOTES:

Pitorifices inserted to center of 4-inch main unless otherwise noted. Values calculated by spreadsheet may differ slightly from original values.

(a) Deflection of water manometer on main apparently used to set flow until it was found not to consistently yield 175 gpm flow.
(b) Kerosene manometer used on service line, divide by 5 to get inches WC.
(c) There was no other reference to a B-3 so this may be B-2.
(d) B-2 was 1 to 1¼ inches before and A-2x1 was ⅜ inch past center.
(e) W-1x1 was inserted 1 inch past and B-2 was ⅝ inch past center.
(f) R-2 was inserted 1 inch past center.
(g) R-1 was inserted 1 inch past center.
(h) This was design selected for project.
(i) Both withdrawn ½ inch from center.

Table A.4: Data and Calculations from Cold Room Tests Done March 27 and 28, 1953

Run	Time	$\dot{m}_s$ lbs/m	gpm	fps	Re	Thermocouple Readings, °F: 1	2	3	4	5	6	7	8	9	10	Water, °F: in 11	out 12	dT	Heat Loss Btu/h	Average Sand Temperature: Dual Pipe	Single Pipe	k_s Btu·in/hr·ft^2·F°
	09:50	11.63																				
	10:06	5.27																				
1	10:15	3.65	0.437	0.294	1120	26.3	32.6	20.4	24.4	22.9	22.9	25.8	34.5	37.2	29.1	39.5	38.1	1.4	306.6	25.2	23.5	4.00
	10:37	-	Moved water reservoir into Cold Room because water temperature was rising 2 F°/hour																			
2	11:17	3.89	0.466	0.313	1113	26.6	31.9	22.8	27.5	25.2	26.0	27.0	33.1	34.4	29.2	35.1	34.9	0.2	46.7	26.9	25.7	0.91
3	12:00	3.62	0.434	0.291	1023	26.8	31.4	24.4	28.5	26.4	26.6	27.7	32.6	33.7	29.1	34.4	34.3	0.1	21.7	27.5	26.6	0.53
4	14:20	3.54	0.424	0.285	959	27.1	29.3	23.8	27.1	25.5	23.4	27.4	30.8	31.5	29.0	32.2	32.1	0.1	21.2	26.3	25.4	0.63
	16:24	3.63	0.434	0.292	988					25.5	21.0	27.5	30.7	31.9	28.8	32.5	32.4	0.1	21.8	25.7		
	19:50	3.53	0.423	0.284	986	27.2	30.5	26.1	28.2	27.1	26.3	27.5	31.7	33.0	28.6	33.7	33.7	0.0	0.0	27.4	27.0	
5	20:10	3.38	0.405	0.272	938	27.3	30.5	26.2	28.3	27.1	26.4	27.9	31.7	32.9	28.7	33.4	33.3	0.1	20.3	27.5	27.1	0.65
6	21:05	3.19	0.382	0.256	872	27.4	30.2	26.3	28.3	27.1	25.3	28.0	31.3	32.3	28.8	32.7	32.6	0.1	19.1	27.3	26.8	0.64
	21:40	3.20	0.384	0.257	873	27.8	29.9	26.4	28.0	27.0	24.0	28.2	31.2	32.1	29.0	32.5	32.5	0.0	0.0	27.1	26.6	
7	08:30	2.62	0.314	0.211	732	28.3	30.5	25.9	28.0	26.6	26.5	27.5	31.4	33.0	29.2	33.9	33.6	0.3	47.1	27.5	27.2	1.54
	08:54	-	Water throttling valve adjusted																			
	08:55	1.11																				
		3.09																				
	09:10	1.87																				
8	11:00	1.73	0.207	0.139	490	28.4	31.8	26.2	28.2	27.0	25.5	27.9	31.8	33.7	29.2	34.8	34.0	0.8	83.1	27.4	27.1	0.18
9	14:30	1.53	0.184	0.123	435	28.6	31.2	27.1	28.8	27.8	26.0	28.2	32.2	34.0	29.4	35.0	34.2	0.8	73.5	27.9	27.6	0.18
	15:00	-	Test terminated and copper tubing removed and inspected																			

NOTES: 1. Data for Table is from the following tables in Page (1953) and Westfall (1953):

a. Tables 7 in both Page and Westfall gave weights of water over intervals of 1.6 to 4.7 minutes.

b. Table 8A in Page gave weights of water over 1 minute intervals

c. Tables 6A in Page and 8 in Westfall gave flows calculated at 1 minute intervals; these were used in their calculations. Page only gave readings for numbered runs; Westfall included other runs.

d. Table 10 in Page gave heat loss rates and thermal conductivities.

e. Page noted that data from Run 1 was not used since temperatures had not stabilized.

2. Values calculated in this spreadsheet may vary slightly from those reported by Page and Westfall.

3. Average sand temperature for single pipe used thermocouples 1, 3, 4, and 6, and those for double pipe used 5, 6, 7, and 10.

APPENDIX B: PITORIFICE PERFORMANCE GRAPHS

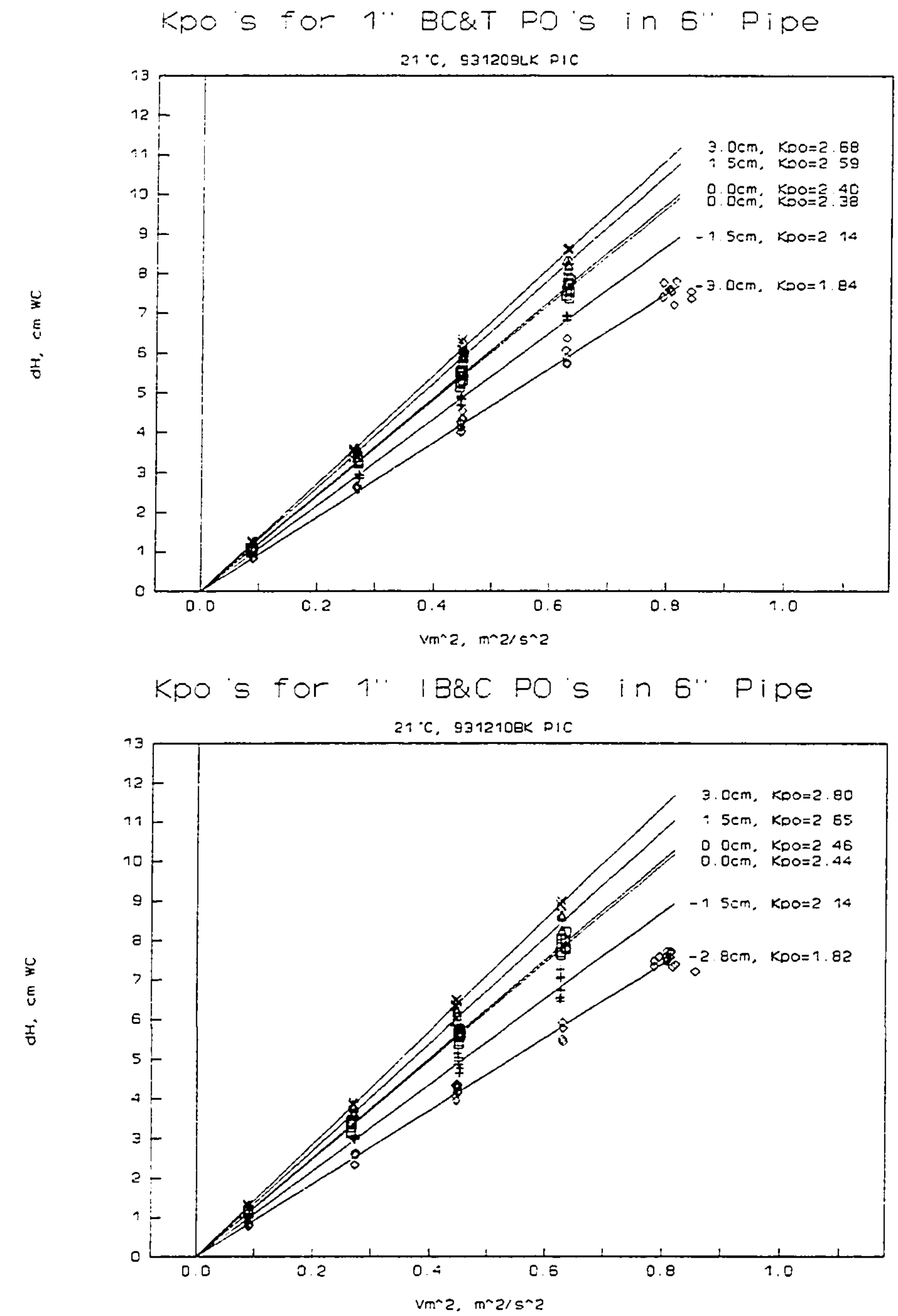


Figure B.1: K_{po}s for 1-inch Pitorifices in 6-inch Pipe

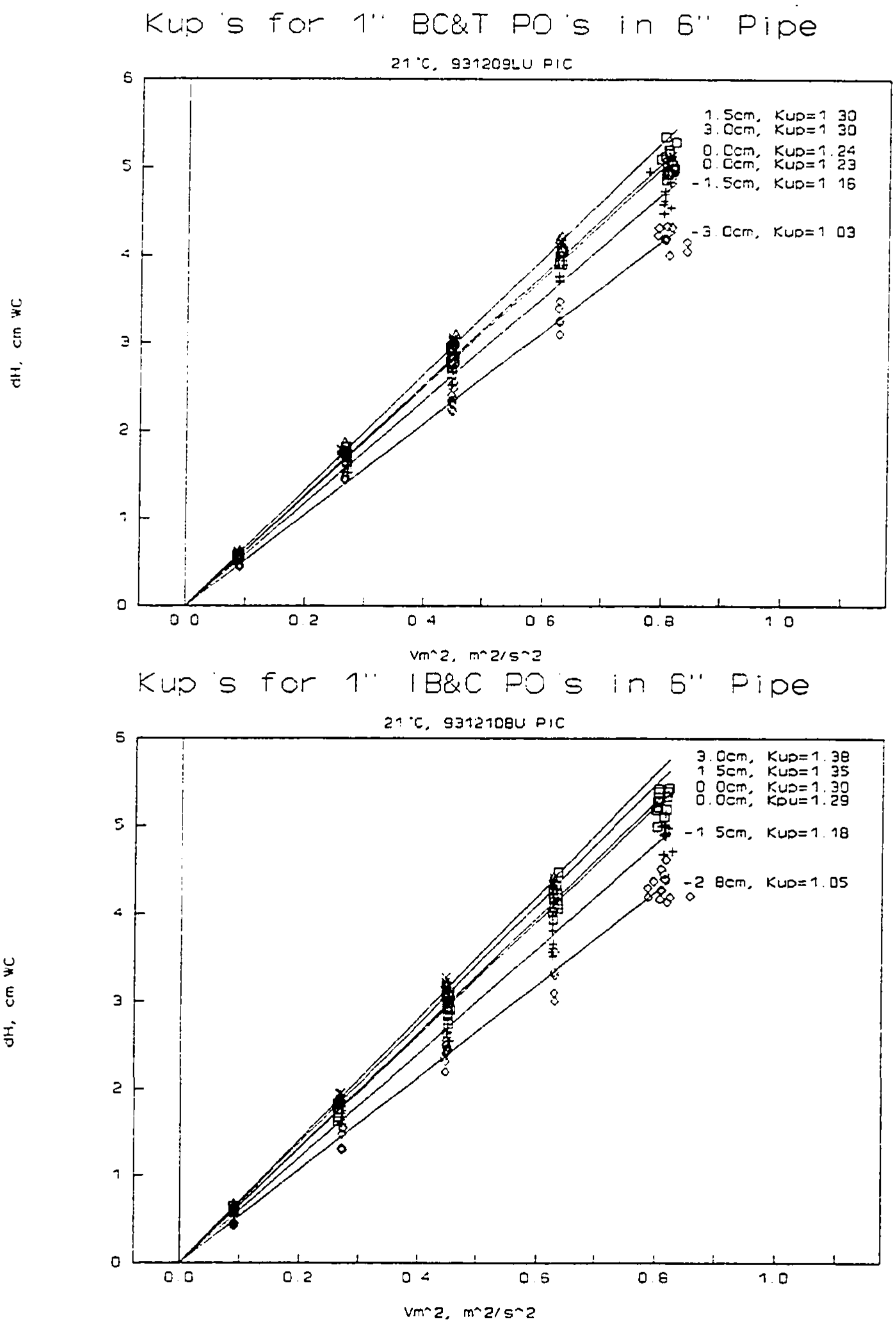
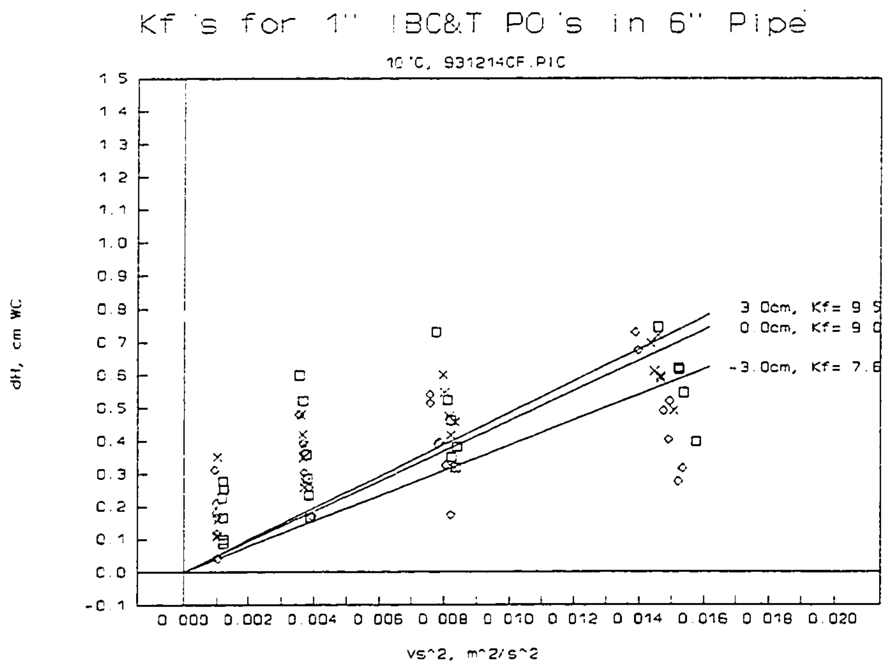


Figure B.2: K_{up}s for 1-inch Pitorifices in 6-inch Pipe



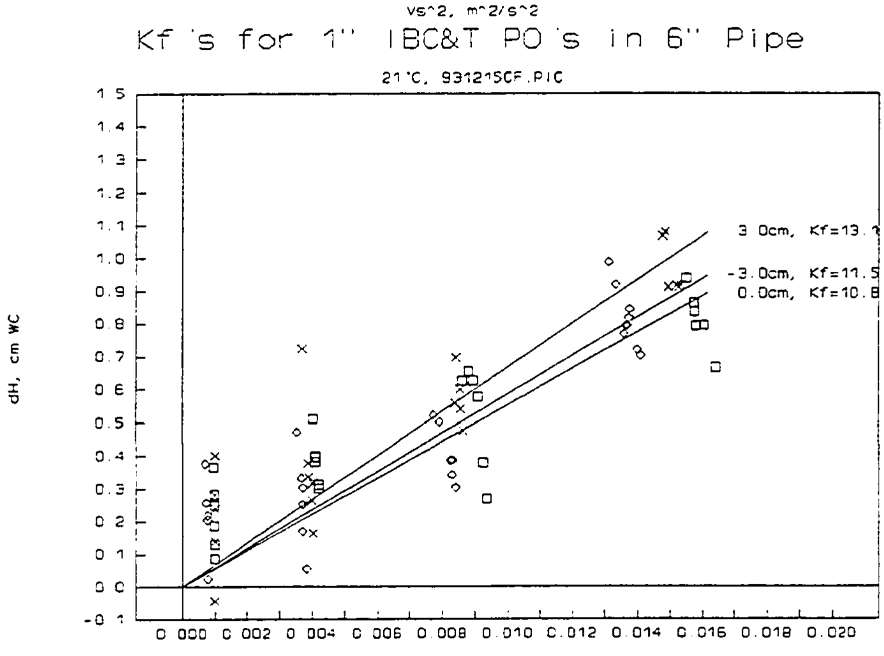
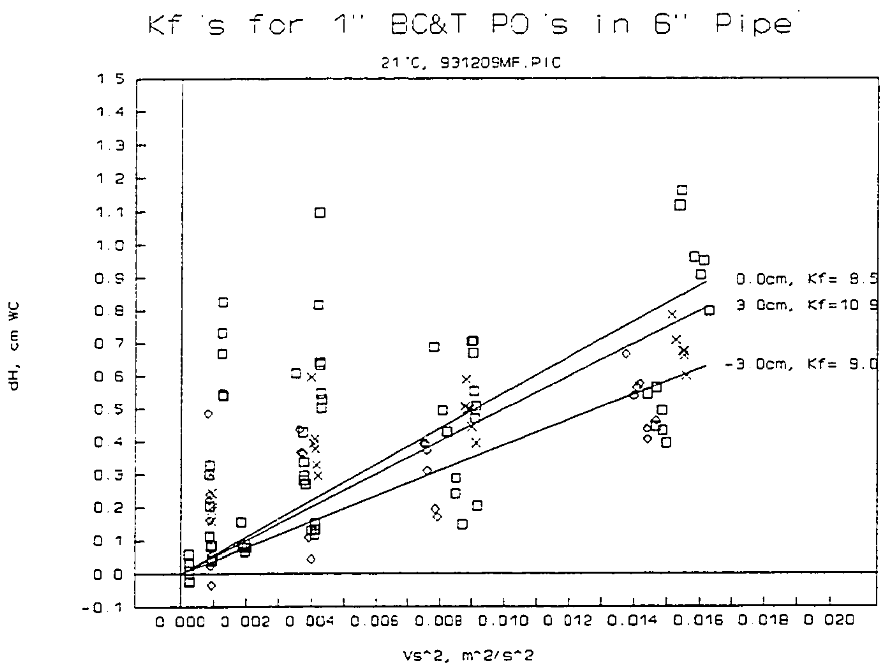


Figure B.3: K_fs for 1-inch Pitorifices in 6-inch Pipe



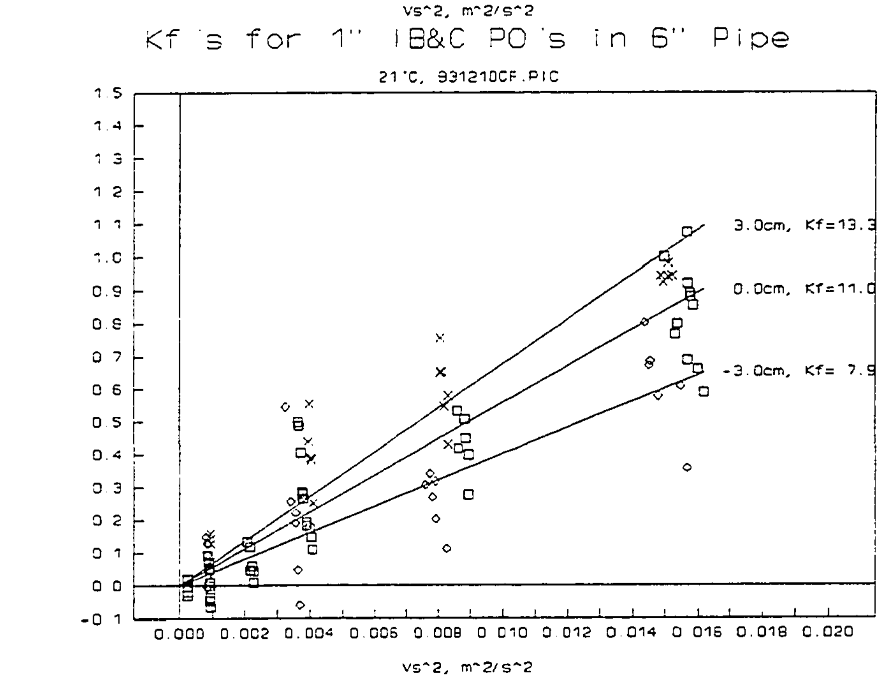
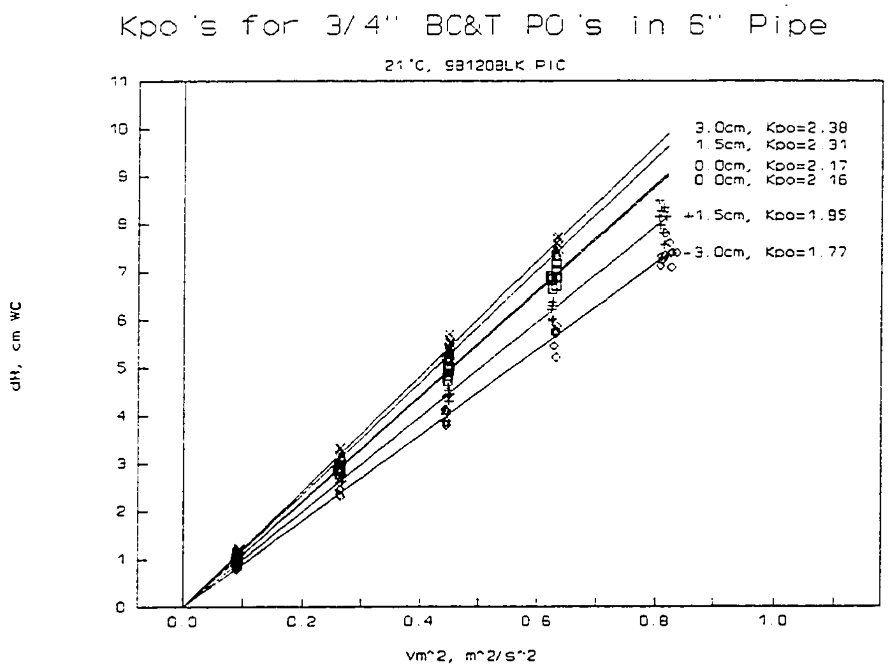


Figure B.3: K_fs for 1-inch Pitorifices in 6-inch Pipe (Continued)



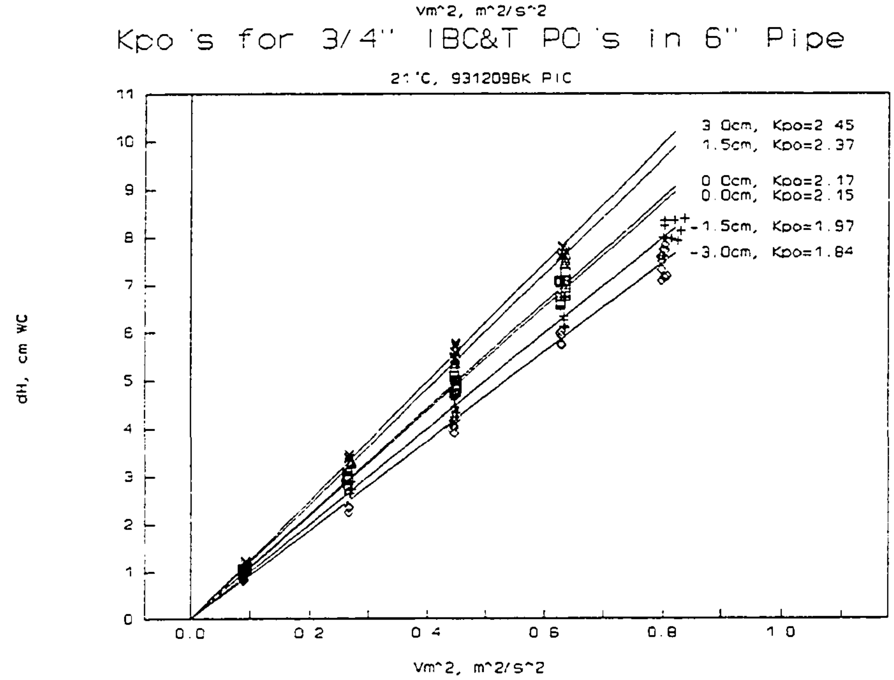
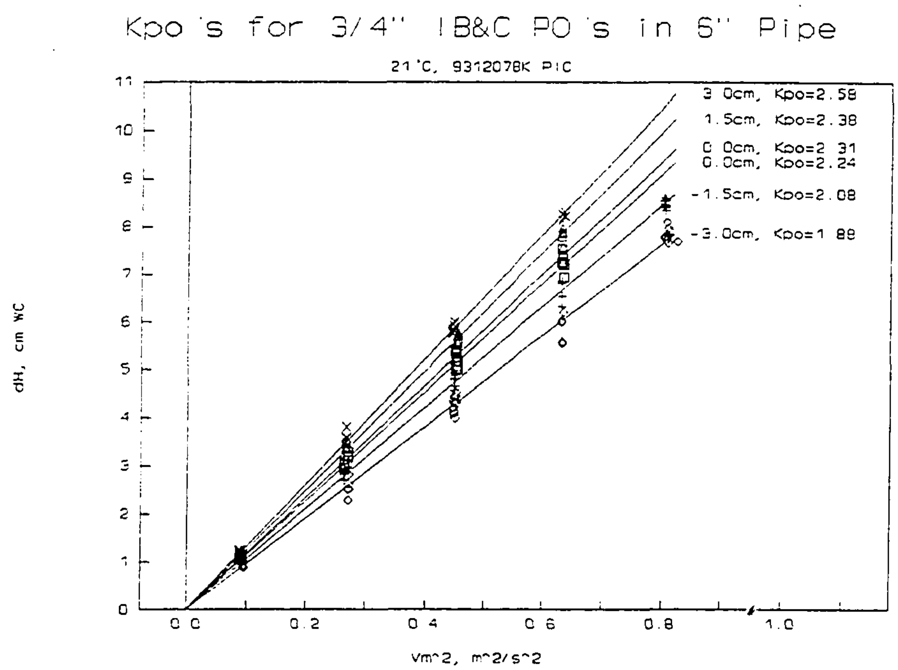


Figure B.4: K_{po}s for 3/4-inch Pitorifices in 6-inch Pipe



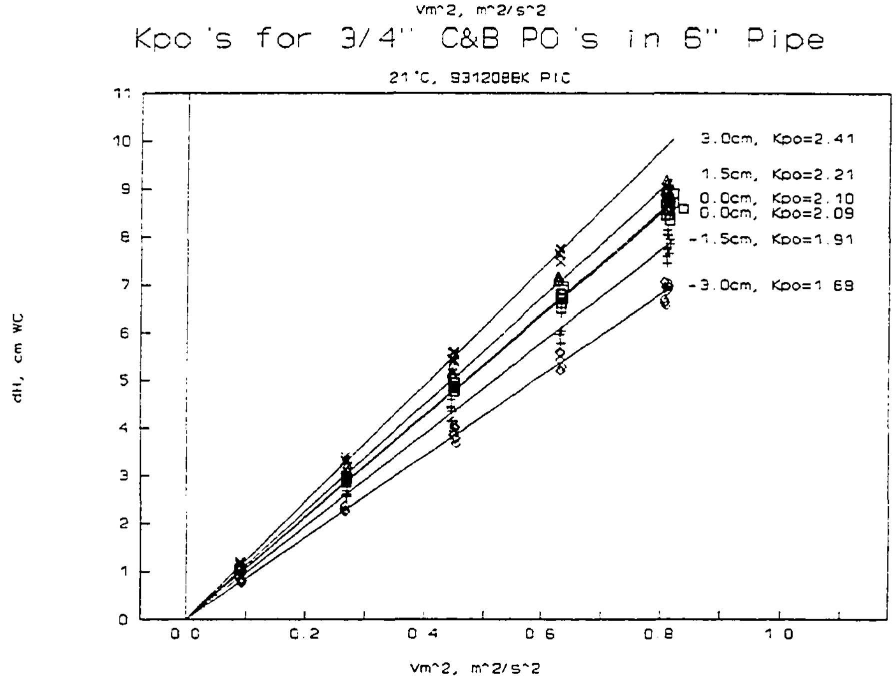
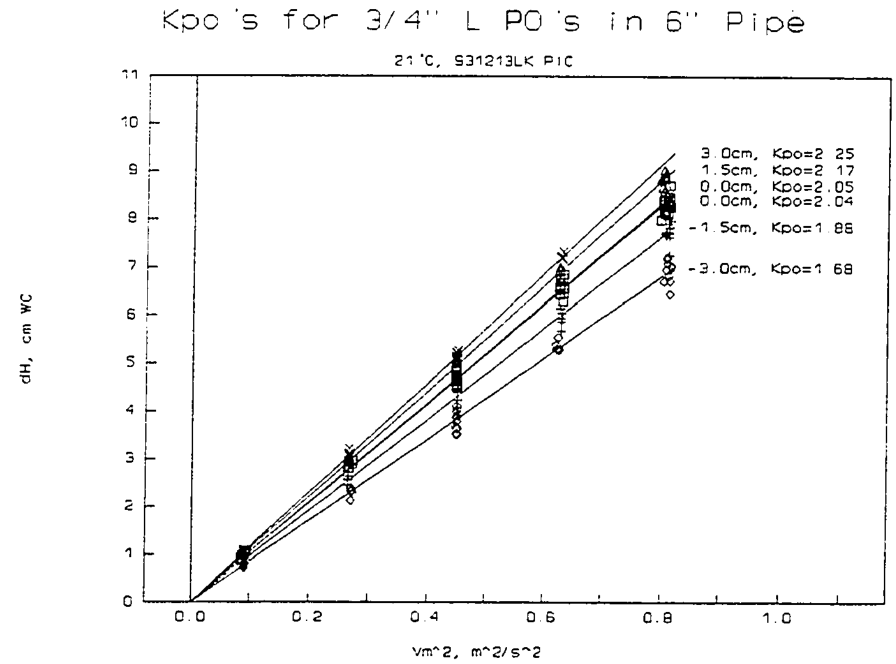


Figure B.4: K_{po}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)



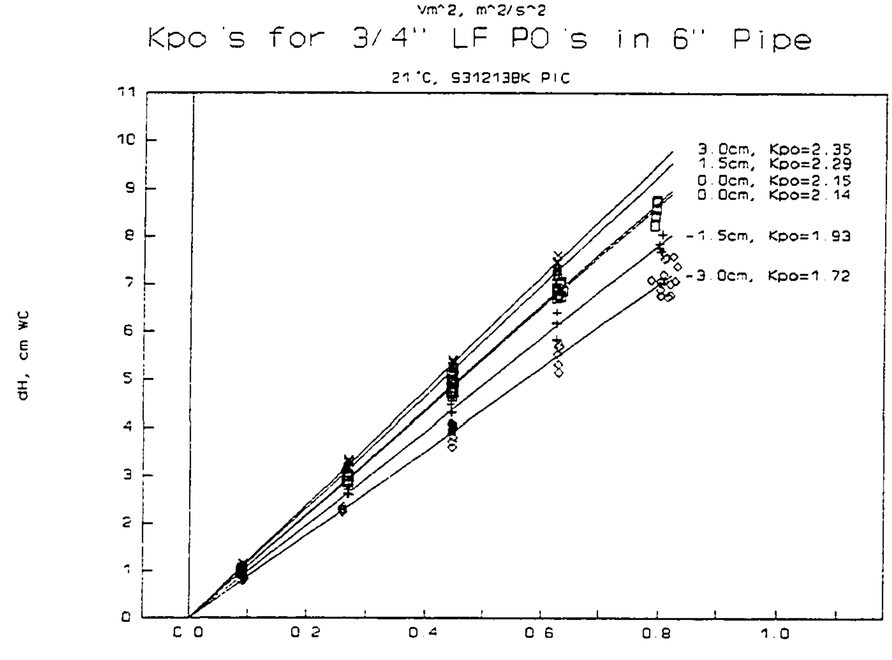


Figure B.4: K_{po}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

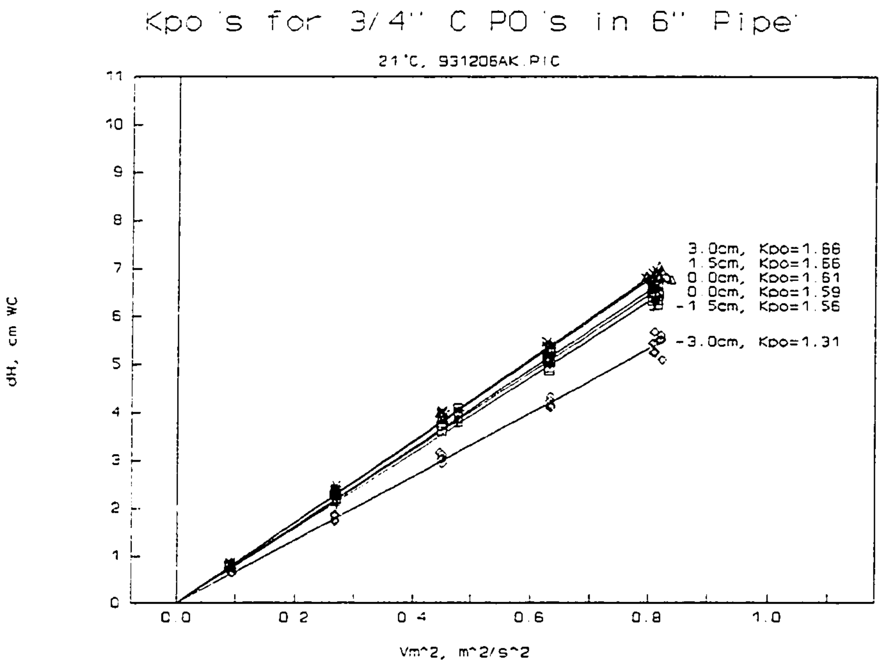


Figure B.4: K_{po}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

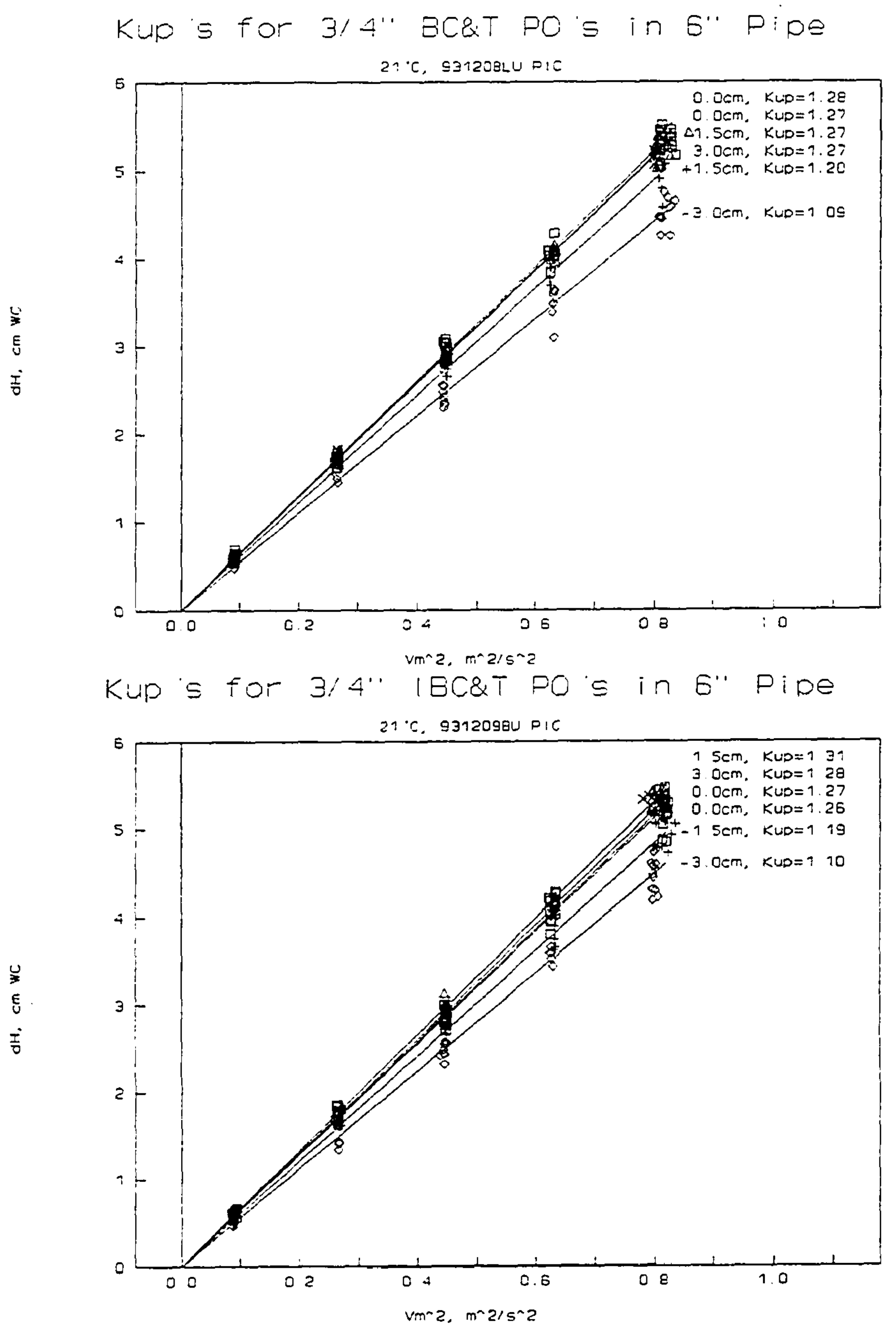
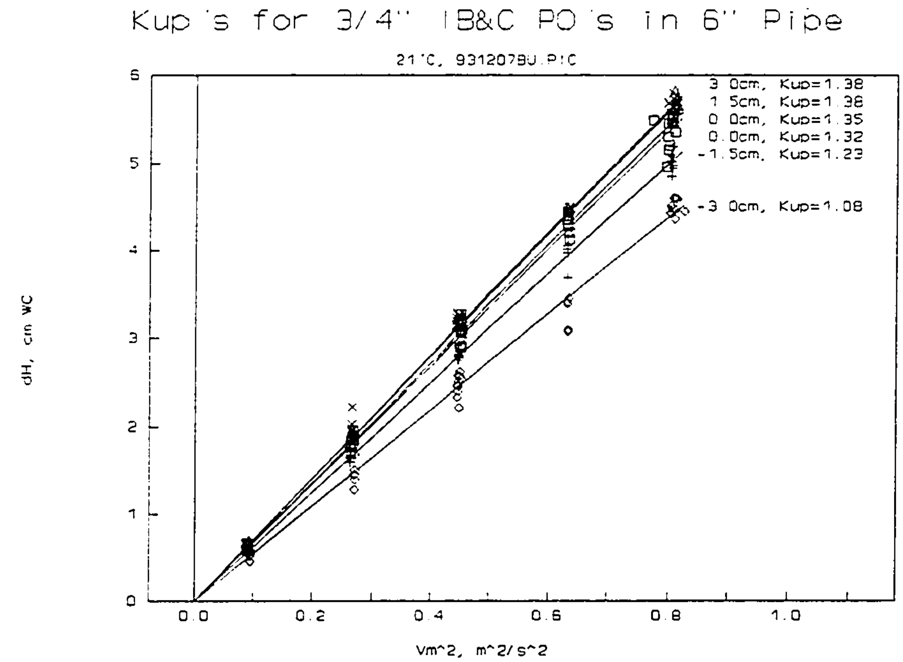


Figure B.5: K_{up}s for 3/4-inch Pitorifices in 6-inch Pipe



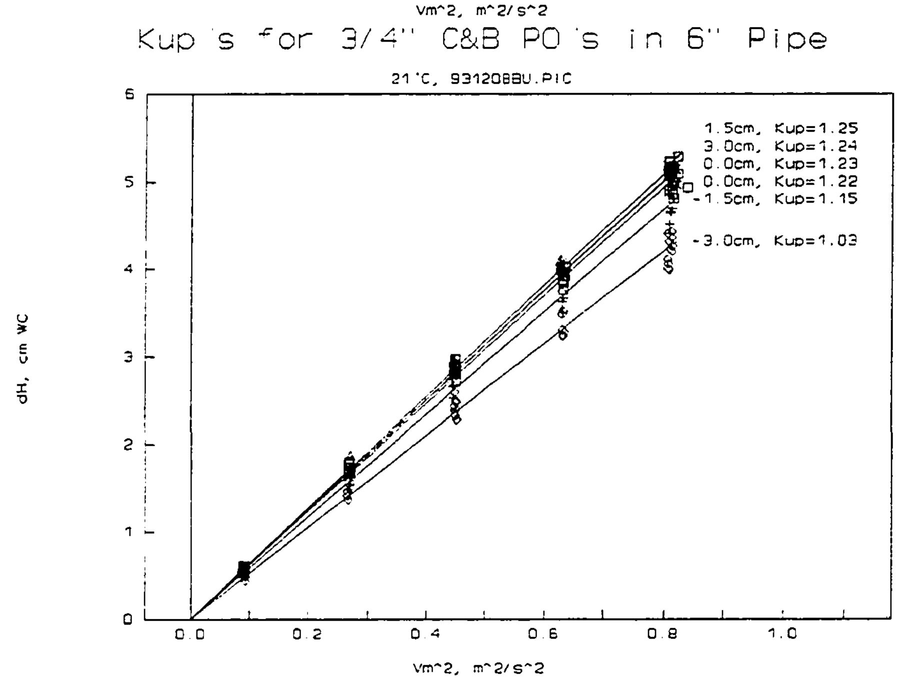


Figure B.5: K_{up}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

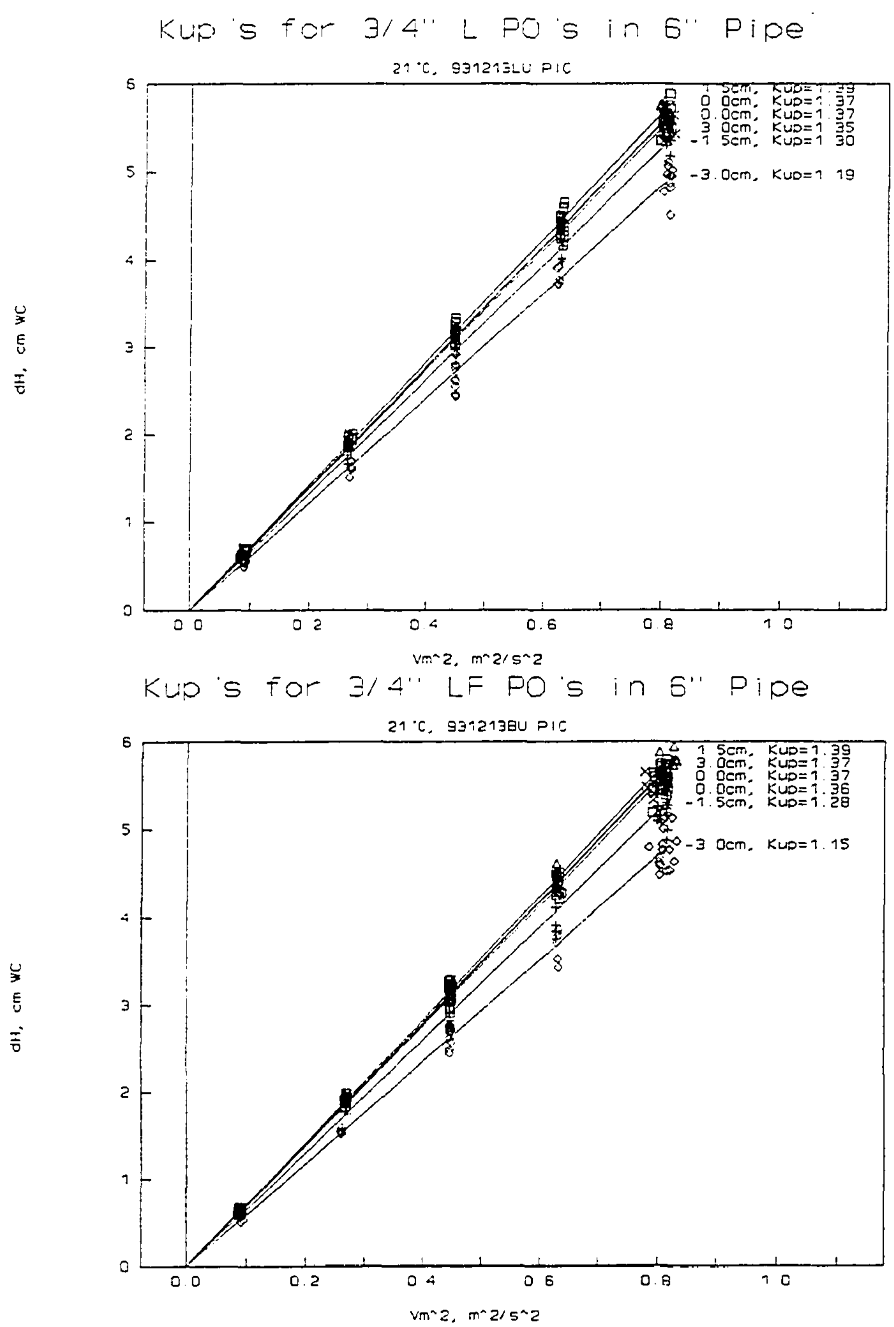


Figure B.5: K_{up}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

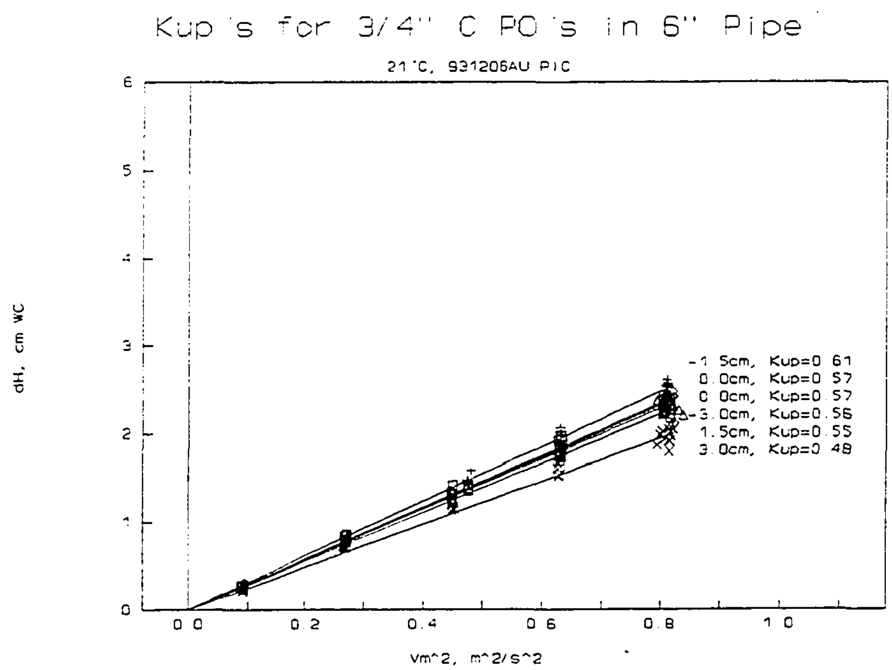


Figure B.5: K_{up}s for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

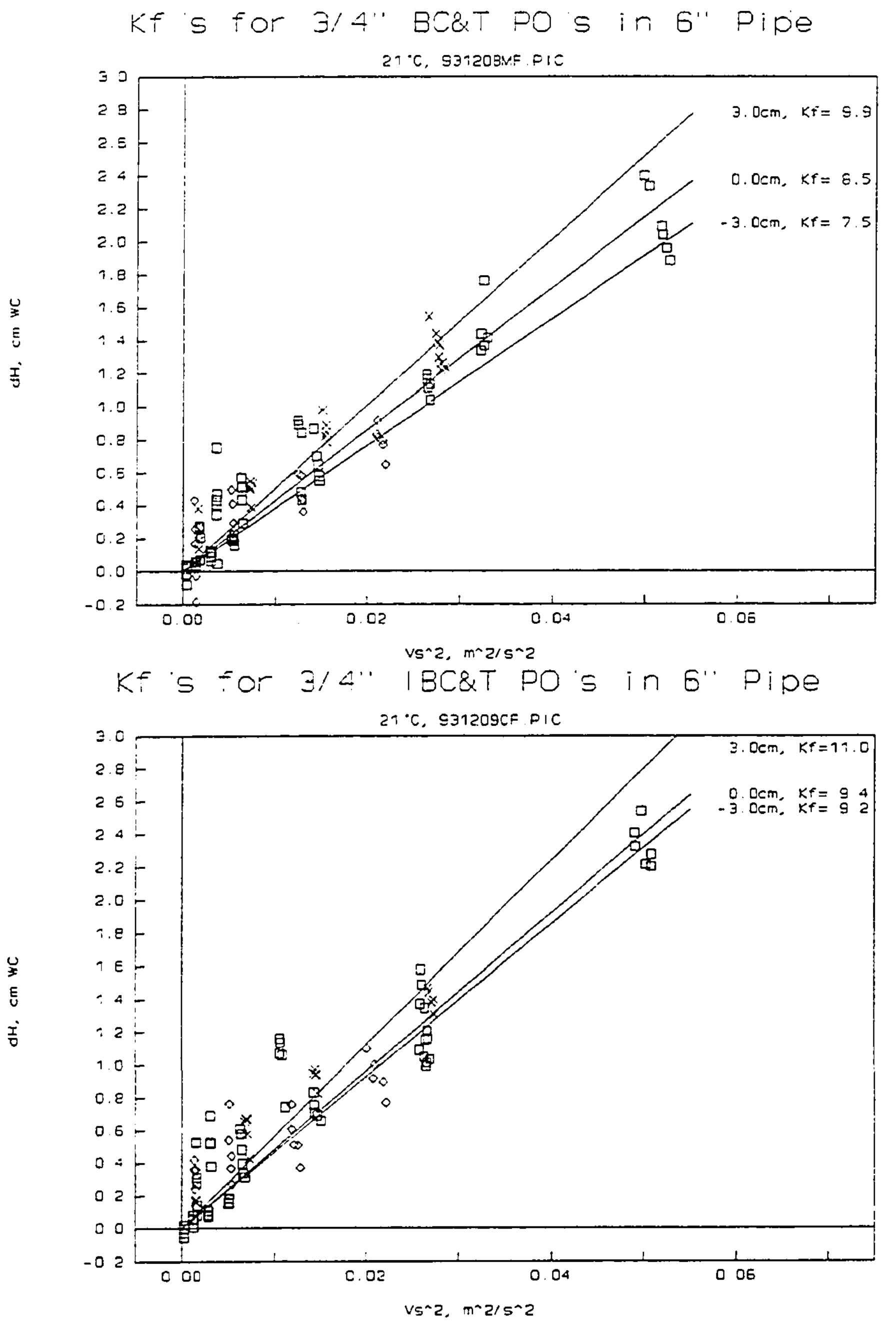
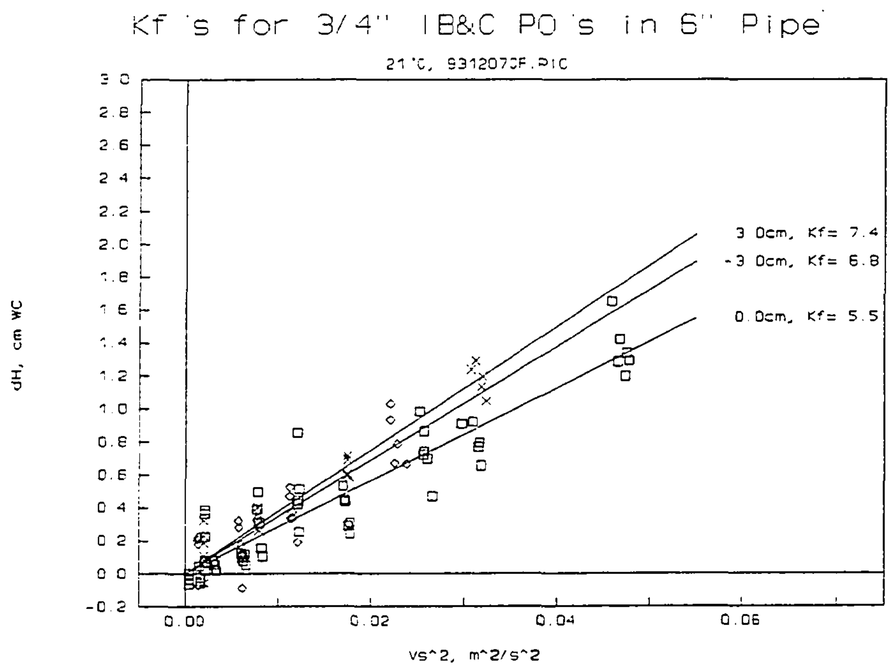


Figure B.6: K_fs for 3/4-inch Pitorifices in 6-inch Pipe



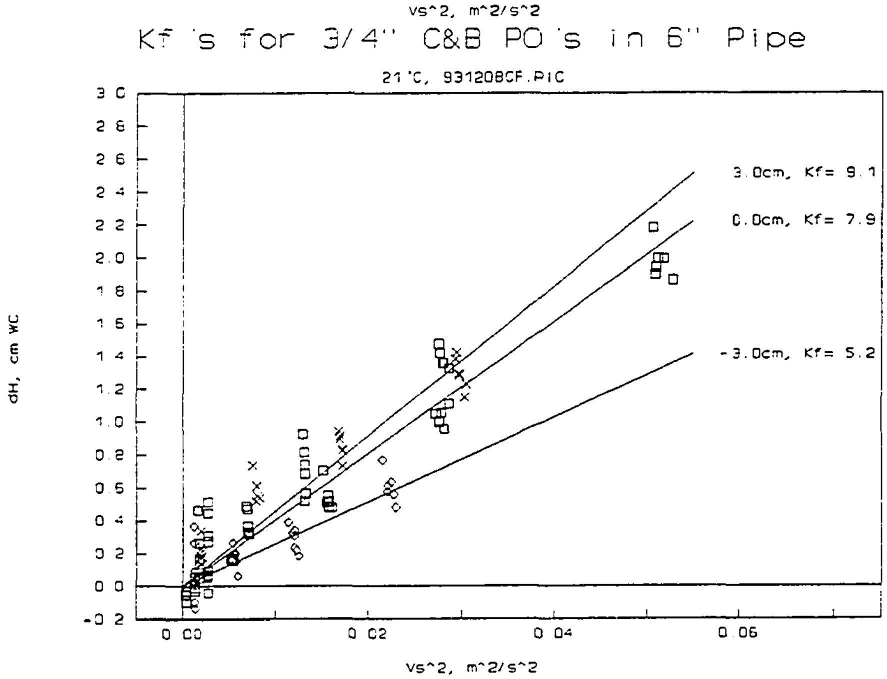
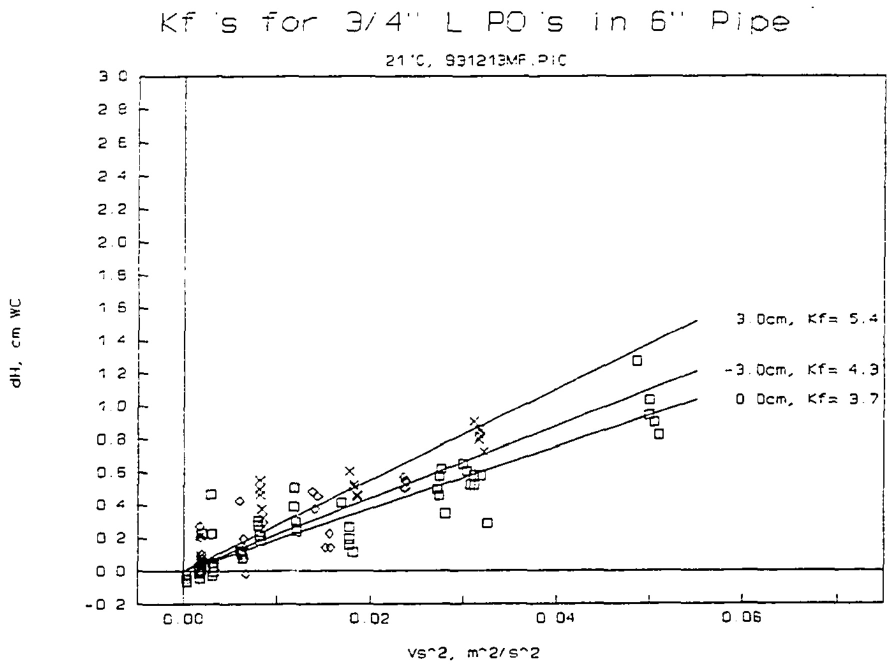


Figure B.6: K_fs for 3/4-inch Pitorifices in 6-inch Pipe (Continued)



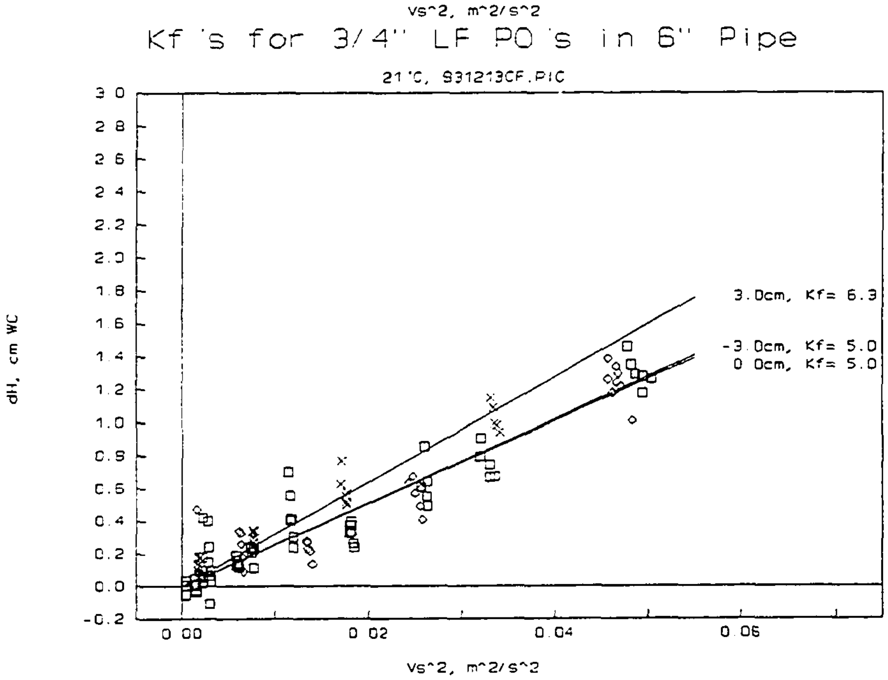


Figure B.6: K_fs for 3/4-inch Pitorifices in 6-inch Pipe (Continued)

APPENDIX C: PITORIFICE DESIGNS

The original design for a pitorifice was up and downstream facing elbows located at the top and bottom of a pipe to exploit a density difference in the water for circulation when there was no flow in the main (Alter, 1950a). Billings (1953) suggested the insertion of an orifice in the main between these "thermal taps", as shown in Figure C.1, would positively make this type of tap operable. This represents the first of many potential and actual modifications to the design of pitorifices.

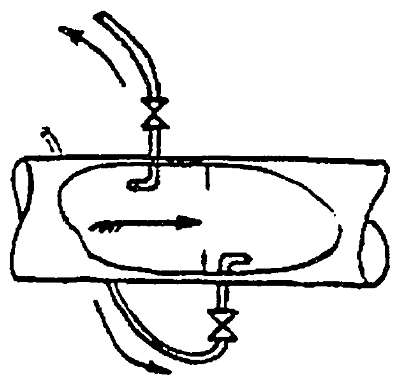
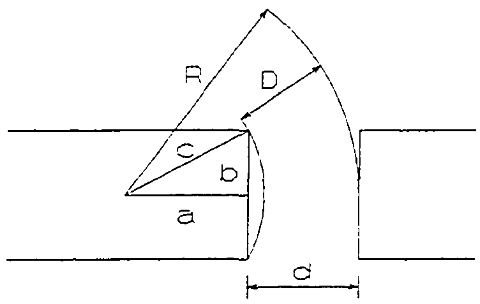


Figure C.1: Orifice Plate in Main Enhances Thermal Tap Performance (Billings, 1953)

The original pitorifice design for Fairbanks in 1953 was a compromise between efficiency and ease of manufacture. It was intended for direct installation in 6-inch ID wood stave mains with 1.12-inch walls reinforced with 8 gauge wire wrapped ¾-inch on-center. Because the wire reinforcement on the wood stave pipe had to be spread for the tap, keeping the hole as small as possible may have been a consideration in choosing a ¾-inch corporation stop which screwed directly into the wood. This in turn led to constructing the pitorifice so that it had a cylindrical profile. Spacing pitorifices a minimum of 8 inches apart may also have been a result of wanting to ensure the intervening piece of stave would not be damaged and break. The hole size necessary to accommodate a pitorifice with a curved profile is given by the following equation and illustrated in Figure C.2, where R is the outside bend radius of the pitorifice, d is the hole diameter, D is the outside diameter of the pitorifice, b is one-half the depth of the hole, and a and c are geometric constructs intermediate to the solution.

The original pitorifices were made with $R \leq 2$ inches and $D \approx 0.81$ inch which means the hole diameter would have to be at least 0.95 inch if the pitorifice had not been turned down in a lathe to a cylindrical profile. A 1-inch diameter hole would have been drilled instead of a $^{13}/_{16}$ or ⅞-inch diameter hole and the threads on the corporation stop would be less tight when it was fully screwed in.



$$\begin{aligned} d &= R - a \\ &= R - \sqrt{c^2 - b^2} \\ &= R - \sqrt{(R-D)^2 - b^2} \end{aligned} \tag{C.1}$$

Figure C.2: Required Hole Size for Bent Pitorifice Clearance

With the subsequent use of pipe saddles on steel, ductile iron, and plastic pipe with a much smaller wall thickness than wood stave and with much less structural reason to limit the hole size or placement, larger pitorifices and bent pitorifices could be considered and they could be placed closer together. Other shapes may be more efficient or cheaper to manufacture or install. The IB&C shape is both more efficient and is cheaper to make than the IBC&T shape. (See Figure 6.1 for shapes.) Alternatively, a more expensive design incorporating two pitorifices into a single tap or allowing wet taps, may allow savings in installation costs.

Some desirable features for a standard pitorifice are low cost, simple to install with symmetry for up and downstream directions to minimize installation errors, low head loss for both the main and the service, high shut-off head development for a given flow in the main, and the ability to shut a line off at the surface. Wet tap or single tap designs would probably cost more but could cut installation costs. Any design will, of course, be a compromise between features.

C.1 REVIEW OF PITORIFICE-LIKE FLOW MEASURING DEVICES

The only devices that resemble pitorifices in form and function are flow measuring devices. Of these, only two types have been identified which actually have water flowing through them. The Bentzel Velocity Tube was developed in 1932 by Carl E. Bentzel for accurate and rapid measurement of very low velocities in streams (ENR, 1934; Everest, 1967; Falkner, 1935). In this instrument water from a stream flowing at 0.15 to 3 fps is diverted by an upstream facing pitot tube past a captive float in a tapered tube (somewhat like an inverted rotameter) and back out a downstream facing pitot tube as shown in Figure C.3. Metraflex (Chicago, Illinois) and Fischer & Porter (Warminster, Pennsylvania) manufacture impact-type rotameters. These consist of an impact tube which is inserted into a pipe and diverts water through a rotameter when a knob is pushed to open a valve, as shown in Figure C.4.

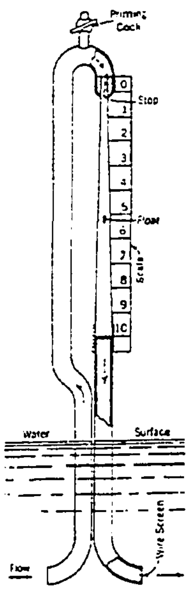


Figure C.3: Bentzel Velocity Tube (ENR, 1934)

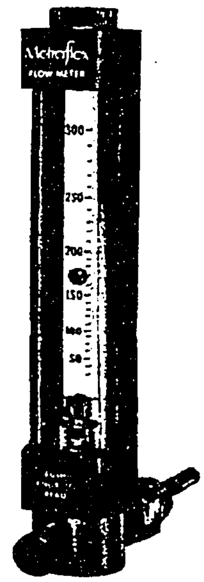


Figure C.4: Impact-type Rotameter (Metraflex, Chicago, Illinois)

There are also numerous devices which measure flow in pipes by noting pressure differences between openings. These can be categorized as uni-directional and bi-directional. A uni-directional wet tap insertion pitot tube is shown in Figure C.5 (Goldstein, 1983). Bi-directional pitot meters as shown in Figure 4.2 have been used for many years for leak detection surveys. Both Dwyer Instruments (Michigan City, Indiana) and Mid-West Instrument (Sterling Heights, Michigan) manufacture averaging pitot meters using tubes (Series DS-200 Flow Sensor and Delta-Tube respectively) such as the one shown in Figure 4.3.

The "static" and downstream openings for both these pitot tubes give a pressure significantly less than true static pressure. The uni-directional design can give a larger differential pressure than a bi-directional design as can be seen in Figure 4.4.

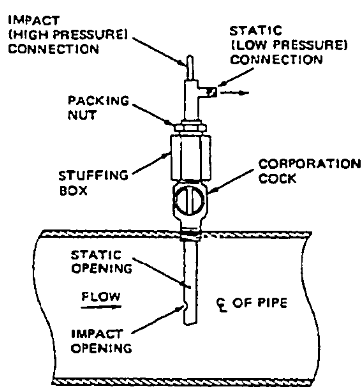


Figure C.5: Uni-directional Wet Tap Pitot Tube (Goldstein, 1983)

C.2 SINGLE TAP DESIGN FOR PITORIFICES

The first and only published depiction of two pitorifices in a single tap is an illustration of how pitorifice systems work reproduced in Figure C.6 (Rice and Alter, 1974). No such tap has ever been used, so the sketch is presumably for illustrating a concept, not a device.

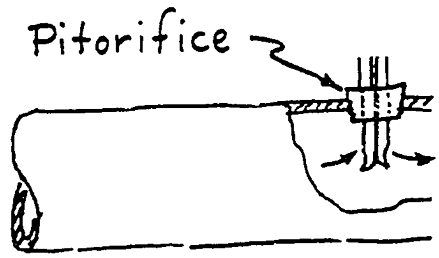


Figure C.6: Pitorifices in Single Tap (Rice and Alter, 1975)

C.3 WET TAP PITORIFICE

A design that would permit a wet tap could also prove very helpful for replacements and retrofits. Many leaks have occurred in Fairbanks where pitorifice pairs were installed so possible future service connections could be made without shutting off a main which entails cross-connection hazards and the need to notify the public. A wet tap design may be expensive but still preferred over the alternatives.

Wet taps have been used since 1871 when Hieronymus Mueller invented and patented the first machine for drilling, tapping and inserting corporation stops in mains under pressure (Mueller Co., 1991). A tapping machine meant for 1 to 2½-inch corporation stops might have enough length in the chamber to accommodate a 1-inch pitorifice but this has never been tried. MUS and CUC prefer to use saddles on the smaller diameter ductile iron mains because of concerns that the wall thickness may not be thick enough to ensure a good seal when tapped. A wet tap pitorifice may have to have a cylindrical cross-section.

C.4 MORE EFFICIENT DESIGN

A Pitot-venturi flow measuring device which could be inserted in a pipe was described by Stoll (1951). Like all pitot devices it has an upstream facing opening for the high pressure or impact pressure tap. Unlike most other pitot tubes the low pressure tap is not designed to measure static or freestream pressure but is in the throat of a venturi nested inside a second venturi. This arrangement allowed differential pressures 5 to 10 times greater than those developed by a pitot element connected to a static pressure tap. Figure C.7 shows both single and double pitot venturi elements (Bean, 1971).

A pitorifice like the Pitot-venturi could be cheaper and easer to install and have less head loss than a flow constriction in the main itself, while still providing the advantages of allowing lower flows in the main than a constriction in the main provides. This device is, unfortunately, uni-directional and expensive.

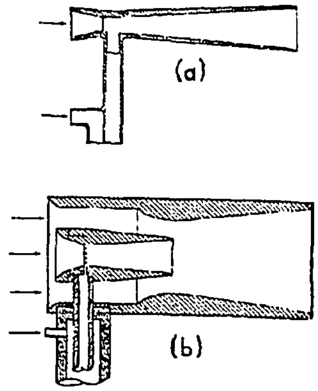


Figure C.7: Pitot-venturi Elements (a) Single (b) Double (Bean, 1971)

C.5 THREAD SIZE AND USE OF SADDLES

Mueller also developed the corporation cock (also called a stop) with special threads which have been referred to as Mueller or CC thread prior to adoption by the American Water Works Association. This thread is listed in ANSI/AWWA Standard C800 as the AWWA Taper thread. This thread has a larger taper than standard pipe thread and the corporation stops with this thread can therefore accommodate a larger pitorifice. A 1-inch corporation stop with an AWWA thread can accommodate a pitorifice made from a nominal 1-inch diameter copper water tube whereas one with pipe thread requires a nominal ⅞-inch diameter copper water tube. AWWA taper thread is preferred by MUS and CUC in Fairbanks because of its increased strength and the lower incidence of breakage compared with standard pipe thread. PHS specifies national standard pipe thread.

Saddles are available with both standard pipe and AWWA taper thread. Standard saddles should not be used on HDPE. A tapped, full wrap-around stainless steel repair clamp and the use of spring washers is used in both Canada and Alaska. Romac Industries (Seattle, Washington), one of the suppliers, will not certify the use of even these saddles on HDPE due to lack of test data.

Saddles should be tightened on cold HDPE pipe and then retightened before backfilling. Freeze-thaw testing of saddles used with HDPE pipe was done by Ortez (1985) following failures in field installations described by Mauser (1984, 1988b). He was unable to cause leakage and concluded that the field installations failed due to inadequate tightening.

The use of tapped repair clamps instead of standard saddles allows double tapping and the placement of pitorifices closer together, however, this has not been done to the author's knowledge. It would allow less bending of the service line to fit it inside a utiliduct and this may be worth the extra care required for hole alignment in the field and the small decrease in pitorifice performance.

C.6 SHUT-OFF AT THE SURFACE

The standard service tap for the Northwest Territories, Canada, consists of saddle taps on the top of the main with a short riser to angle ball valves followed by a horizontal run to the house. The riser and angle ball valves are contained within an insulated enclosure. Capped valve access pipes are terminated below grade. About 3 inches of expanded polystyrene insulation with a ½-inch plywood cover is placed above the valve access for freeze protection (Cheema, 1986; Wilson and Cheema, 1989).

Mathews (1992) designed a similar system for Marshall, Alaska, using pitorifices epoxied into ball valve curb stops installed horiztally in the main with 2-inch diameter HDPE valve actuator risers to the surface for access. Epoxy was used because the pitorifices could not be soldered into the valve body without removing the ball valve seals. The seals could have been removed and the valve reassembled after soldering but the machined pitorifice opening would have been difficult to seal for pressure testing. Standard size tubing could have been used had AWWA threaded corporation stops been used. If standard size tubing was used and the pitorifice shape was to be a bend and cut type (the most efficient and in fact the shape used in Marshall), then the tubing could have been bent, capped, and soldered into the body of the valve. The valve could then have been assembled, tested, and the end of the tubing cut off.

A problem with any system of accessing the service at the main is that a heat can be lost through the access pipe and rods which can lead to freezing. Also, water can be trapped inside a closed corporation stop or ball valve, and when this water freezes the valve can break. Drilling a weep hole in the body of the valve is not recommended because of the potential for contamination. Finally, soil and debris can fill an open pipe access for a rod and prevent operation. This can be due to misalignment from settling or heaving, through damage to the cover, or from children or vandals. Filling the annulus with insulation can help prevent problems of this sort.

C.7 ALL HDPE TAP

With the use of HDPE mains and service lines has come an interest in eliminating saddles and even fittings and using a fusion weld tap. A prototype of a HDPE spool with two pitorifices inserted in side-fusion taps was made by Maskell-Robbins (Anchorage, Alaska) for the 1995 Bethel FAA housing complex design by FPE Roen Engineers, Inc. (Fairbanks, Alaska). A section of 6-inch diameter HDPE pipe was fitted with two standard nominal 1-inch diameter IPS SDR 9.3 side fusion taps 3.5 inches apart. The taps were bored with a $^{15}/_{16}$-inch diameter drill and IBC&T type pitorifices made from 1.000-inch OD copper water tube inserted and locked into position by deforming the end of the copper tube inside the tap. This design is a modification of a design for prefabricated water stub-outs described by Thomas (1988) for use in Barrow, Alaska, where distributed pumping is used.

Destructive testing of factory fusion welds in HDPE and certification of welders using destructive testing is strongly recommended because a poor weld may not leak until after several years of service. Long term testing of HDPE pipe and fusion weld specimens supplied by three different manufacturers resulted in the unexpected discovery of numerous defective factory fusion welds (Mauser, 1985; Mauser, 1987a; Mauser 1987b). The testing was performed by the Springborn Testing Institute in accordance with ASTM D 1598-81 Standard Test Method for Time-to-Failure of Plastic Pipe Under Constant Internal Pressure.

Random fittings with factory welds should be tested by cutting the pipe with the weld into 1-inch wide strips and bending back and forth until there is a failure in the pipe or the weld (Mauser, 1988a). If any of the strips fail in the weld the piece should be considered to have failed and all associated pieces should be rejected. Field welders should be trained and tested in a similar fashion on all types of welds they will be making.

APPENDIX D: PUMP AND MOTOR EFFICIENCIES

The following sections discuss the derivation of empirical relationships used in Chapter 10 relating pump and motor efficiencies as a function of capacity. Other items of concern to the engineer working with cold water or small pumps are also discussed.

D.1 PUMP EFFICIENCIES

Sanks (1989) presented a curve showing maximum efficiency of centrifugal pumps as a function of capacity. Figure D.1 and Equation D.1 below were obtained fitting an exponential curve through points on Sanks' curve. Values from the curve can be considered the typical maximum efficiency when a pump has a specific speed of around N_s=3,000.

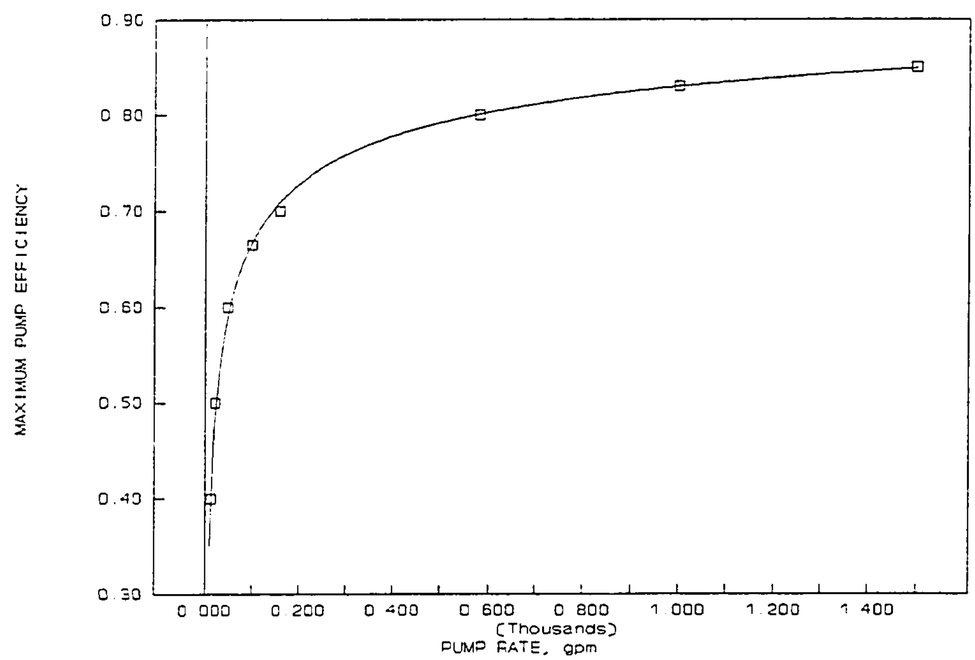


Figure D.1: Maximum Pump Efficiencies

$$\eta_p = 1 - 1.27(Flow\ in\ gpm)^{-0.291} \qquad \textbf{(D.1)}$$

Specific speed, n_s, is a dimensionless number used to characterize pump performance at the best efficiency point (BEP):

$$n_s = \frac{nQ^{0.5}}{(g\,\Delta H)^{0.75}} \qquad \text{(D.2)}$$

where n is the rate of revolution, Q is the volumetric flow rate at the BEP, g is the acceleration of gravity, and ΔH is the head at the BEP. In the United States, specific speed is customarily defined with inconsistent units:

$$N_s = \frac{(pump\ rpm)(pump\ gpm)^{0.5}}{(pump\ head\ in\ feet)^{0.75}} \qquad \text{(D.3)}$$

so that $N_s = 17{,}182 n_s$. The best efficiencies for centrifugal pumps are usually obtained with a specific speed of about $N_s = 3{,}000$ [$n_s = 0.17$] as shown in Figure D.2 from White (1986). A pump with an 80 gpm flow rate at 10 feet of head and 1750 rpm would have a specific speed of $N_s = 2{,}800$. From Figure D.2 it can be seen that the efficiency would likely be less than 70 percent (Equation D.1 predicts an efficiency of 65 percent).

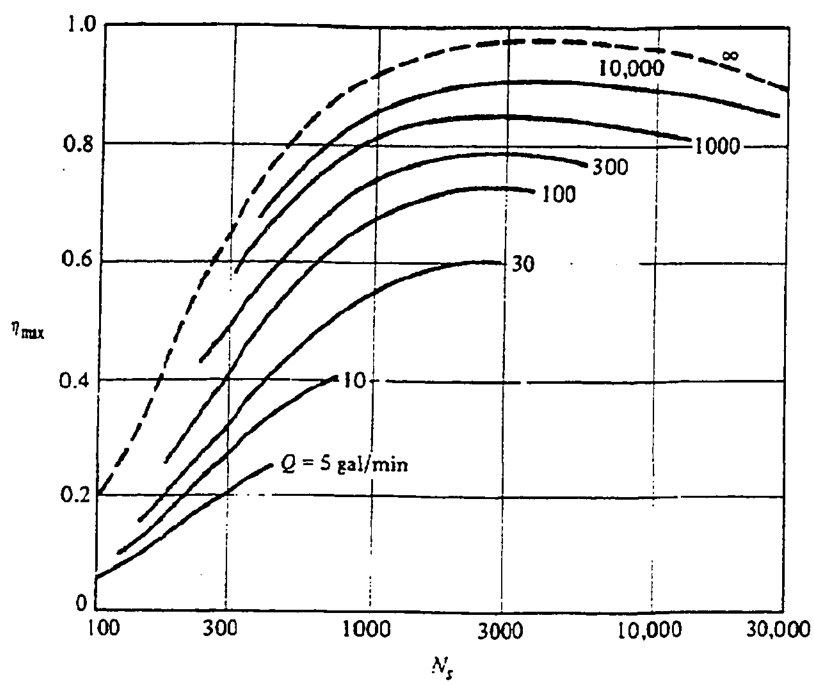


Figure D.2: Maximum Efficiency Versus Specific Speed (White, 1986)

The efficiency of a pump will be affected by the viscosity of the fluid. While a lower viscosity will result in lower friction losses it will also result in higher internal leakage. The net effect will depend on the specific speed and design details of the pump. The Hydraulic Institute (1975) gives the following relation between kinematic viscosities and efficiencies at pump test and operating conditions:

$$\eta_o = 1 - (1 - \eta_t)\left[\frac{\nu_o}{\nu_t}\right]^n \qquad \textbf{(D.4)}$$

The exponent, n, typically ranges from 0.05 to 0.1 and must be established from manufacturers' data. Assuming n=0.1 and an efficiency of 80 percent at a test temperature of 85°F, the efficiency should be 78.5 percent at 35°F for a loss of less than two percent.

D.2 MOTOR EFFICIENCIES

Figure D.3 and Equation D.4 below were obtained by plotting and curve fitting the information given by Reliance Electric (1991) for standard three-phase 1800 RPM normal service industry average and energy efficient motors. Values for single phase energy efficient motors are given in Table D.1. The motor efficiencies are determined at full load conditions. Motor efficiency for standard NEMA frame motors reaches its maximum at a point below its full rated load (Reliance Electric, 1982). Since pump motors are usually not fully loaded, it is generally safe to use reported efficiency values.

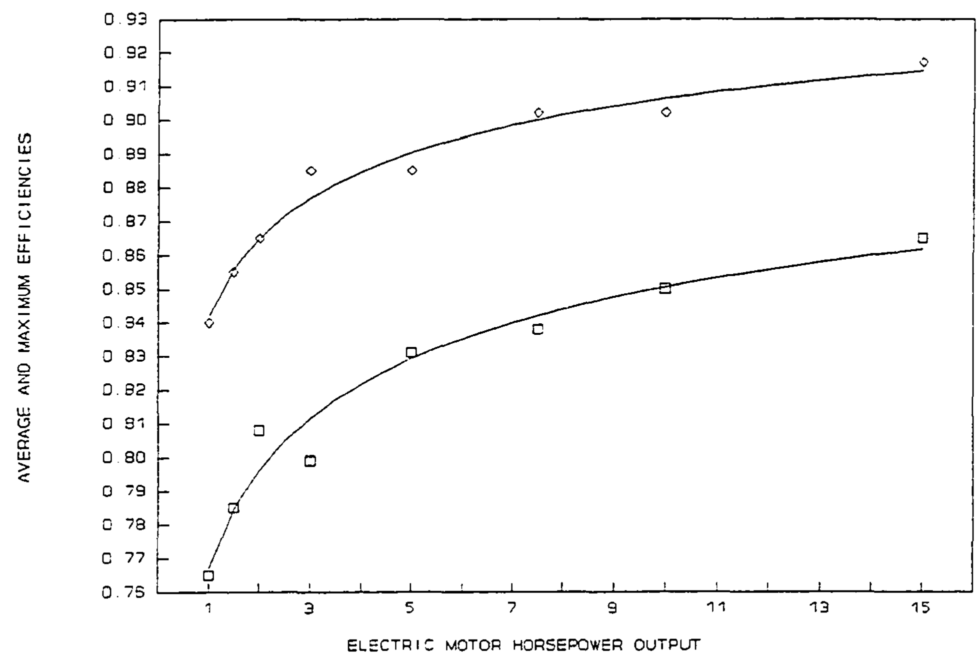


Figure D.3: Average and Maximum Electric Motor Efficiencies

$$\eta_{avg} = 1 - 0.233(HP)^{-0.193}$$
$$\eta_{max} = 1 - 0.158(HP)^{-0.227} \qquad \textbf{(D.5)}$$

Table D.1: Performance of Energy Efficient, 1,800 RPM Motors

Horse power	¼	⅓	½	¾	1	1½
Efficiency	0.70	0.72	0.77	0.77	0.77	0.80

There can be a cost benefit in purchasing an energy efficient motor for a pump that is run for three months of the year. Assume a general purpose three phase 2 Hp motor costs $70 more for the energy efficient model with an efficiency of 86.5 percent instead of only 80.8 percent. With 70 percent loading, the same efficiency at 70 percent loading as at 100 percent loading, electricity at $0.10/Kwh, and 90 days per year operation (two pumps are often used with one on standby at all times) the annual savings from the energy efficient motor will be a little over $18 resulting in a four year payback.

D.3 SMALL CIRCULATING PUMPS

The efficiency of small circulating pumps is generally not given in the manufacturers' literature, but the wire-to-water efficiency, $\eta_{w\text{-}w}$, may be calculated from power requirements and pump performance curves. For example, a Grundfos UPS 20-42 set on its lowest speed will pump 4 gpm at 3 feet WC and use 50 watts yielding an overall efficiency less than 5 percent:

$$\eta_{w-w} = \frac{Q\,\Delta H}{3960\,gpm{\bullet}ft{\bullet}Hp^{-1}} \times \frac{745.7\text{ Watts/Hp}}{(\text{Power in Watts})} = 4.5\,\% \tag{D.6}$$

D.4 POWER FACTORS

The power factor is the cosine of the phase angle between AC current and voltage wave forms. It can also be considered the ratio of real or active power to apparent power:

$$\cos\phi = P/EI$$
$$\text{Power Factor} = \frac{\text{Real Power}}{\text{Apparent Power}} \tag{D.7}$$

Apparent power is the product of root mean square (rms) voltage and amperage. If a consumer has equipment which causes the voltage and current to be significantly out of phase, the power factor will be significantly less than unity. Although the actual power used may be the same as that from equipment with a power factor of unity, the current and hence the line losses will be higher.

The power factor must be used when calculating wattage from AC voltage and current readings.

Alternating current electric motors are either synchronous and have a power factor of unity or they are induction and have a power factor which can vary from 0.2 to 0.9 over the range of no load to overloaded. Induction motors are most commonly used and their power factor is typically around 0.8 under normal operation.

APPENDIX E: AIR LOCKS IN SERVICE LINES

In the following discussion the term "air" can mean any gas or mixture of gases that can be dissolved in and released from water. No cases of service line freezing due to air locks were positively identified. Normal water use by residents probably serves to vent air faster than it can build up. Air locks may be the reason lines to some vacant houses or houses with very poorly insulated service lines froze.

Air locks can prevent circulation in service lines which use either individual circulation pumps or pitorifices. Air can be introduced from taps if a system is partially drained, it can be introduced from a well pump or a pressure tank, or it can be released from water when pressure is relieved or the temperature raised. If the quantity of air is sufficient to prevent the circulation pump or pitorifices from working, service line circulation will stop and freezing may occur.

E.1 TYPES OF SYSTEMS THAT MAY BE MOST SUSCEPTIBLE TO AIR LOCKS

A water system with one or more of the features shown in Figure E.1 may be susceptible to air locks from introduced air or from release of dissolved air. Air may be pumped from the well with the water under some circumstances. This entrained air may dissolve and saturate the water or be conveyed as bubbles to the service lines. Alternatively, the water in the well may be under considerable head and

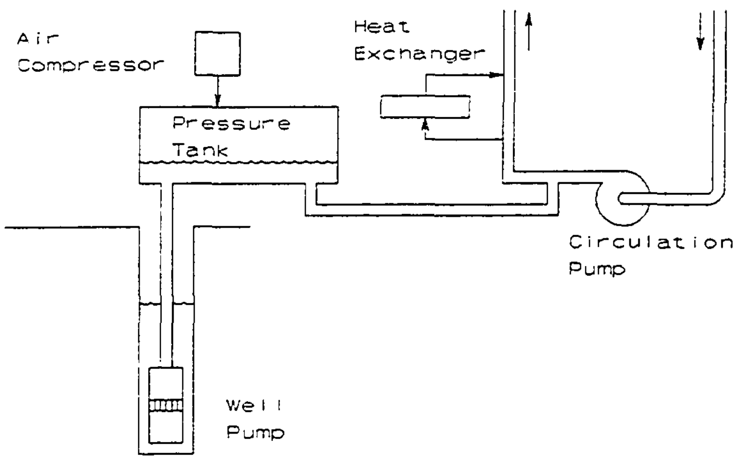


Figure E.1: Schematic of Air Lock Prone System

already contain significant amounts of dissolved gasses. Another source of air is the pressure tank, which will introduce air to the system if the level is not properly maintained or if there is a failure in the supply from the well. The pressure tank can also contribute dissolved air to the water. There are no air scoops or other traps shown in the water plant schematic. Portions of the distribution system shown in the schematic are higher than the water plant, so air will be trapped in mains where it can be dissolved or conveyed to service lines. Finally, the water pressures in the system are lower than those in the water plant so that entrained bubbles will grow in size and gasses may be released from solution.

Proper location of the well pump, use of bladder type pressure tanks, heating make-up water, trapping released air before adding makeup water to the distribution system lines, and air traps and vents at high points will help alleviate air lock problems. Adding make-up water upstream of the circulation pump might also help but is not recommended because it requires additional circulation pump capacity and pumping costs.

The least susceptible water system is one where water pressure is maintained by an elevated pressure tank. In such a system, water in the system is always under more pressure than the water in the tank. The possibilities of either air being introduced into the system or air being released from the water are remote.

E.2 MINIMUM AIR NEEDED TO CAUSE AN AIR LOCK

The minimum air required to air lock a circulation pump will depend on the pump, how much and how fast air is introduced if the pump is running, and on whether the pump is running when air is introduced. The larger the pump, the greater the amount of air it can hold before air locking. The faster the flow of water through the pump, the less likely it is that air will accumulate at all. More air may be required to air lock a distributed pumping system than a pitorifice system particularly if the pumps are not located at the high points.

Assuming that there is a 1.5-inch WC head from a pitorifice pair and that the high point in a service line is as shown in Figure E.2 so that the length of the air column is 4.5 inches, then the volume of air in cubic inches which is required for an air lock is 4.5A where A is the cross sectional area of the line in square inches.

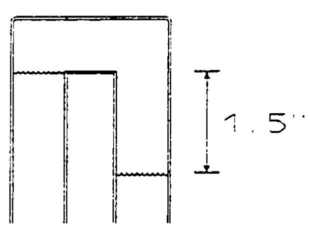


Figure E.2: Air Lock in Service Line with Pitorifices

E.3 AMOUNT OF AIR IN SATURATED WATER

The amount of air that will dissolve in water depends on the composition of the air, and the temperature and pressure of the water. Oxygen (O_2), nitrogen (N_2), argon (Ar), and carbon dioxide (CO_2), make up 20.946, 78.084, 0.934, and 0.032 percent of air. For estimating purposes it may be assumed that the air that initially saturates the water is 22 percent O_2 and 78 percent N_2. It should be kept in mind, however, that CO_2 is very soluble and may be present in significant amounts in well water.

The saturation concentration of a gas in water is given by:

$$C_i = \frac{(P_T - P_{H_2O})}{760\ mm\ Hg} \beta_i X_i \tag{E.1}$$

where C_i is the liters of gas i at standard temperature and pressure (STP) of 0°C and 760 mm mercury column (Hg), P is the pressure in mm Hg with the subscripts T and H_2O referring to total and water vapor, β_i is the Bunsen coefficient of gas i, and X_i is the mole fraction of gas i. Table E.1 gives the partial pressure of water vapor and the Bunsen coefficients for atmospheric gases at temperatures of 4 and 15°C. Additional information may be found in Colt (1984).

Table E.1: Partial Pressure of Water and Bunsen Coefficients for Gasses

	4°C	15°C
P_{H2O}	6.10 mm Hg	12.79 mm Hg
β_{O2}	0.04413	0.03423
β_{N2}	0.02150	0.01704
β_{Ar}	0.04829	0.03759
β_{CO2}	1.4805	1.0136

E.4 AIR RELEASED WHEN PRESSURE IS DROPPED

Assuming air with 22 percent O_2 and 78 percent N_2 saturates water at 4 atmospheres of total pressure and 4°C, the gas concentrations in the water predicted by Equation E.1 will be:

$$\begin{aligned} C_{O_2}^* &= 0.0388\ \textit{Liters per liter of water} \\ C_{N_2}^* &= 0.0669\ \textit{Liters per liter of water} \end{aligned} \tag{E.2}$$

If this saturated water is now reduced in pressure to 1 atmosphere of total pressure, gas concentrations at equilibrium will be:

$$\begin{aligned} C_{O_2} &= 0.0438 X_{O_2} \\ C_{N_2} &= 0.0213 X_{N_2} \end{aligned} \tag{E.3}$$

Using V_T for the total volume of released gasses at STP per liter of water, we can write:

$$C_{O_2}^* - C_{O_2} = X_{O_2} V_T$$
$$C_{N_2}^* - C_{N_2} = X_{N_2} V_T \qquad \text{(E.4)}$$

Noting that:

$$X_{O_2} + X_{N_2} = 1 \qquad \text{(E.5)}$$

results in five equations and five unknowns. V_T can be solved for by substitution:

$$0.0388 - 0.0438 X_{O_2} = X_{O_2} V_T$$
$$0.0669 - 0.0213 X_{N_2} = X_{N_2} V_T \qquad \text{(E.6)}$$

$$X_{O_2} = \frac{0.0388}{V_T + 0.0438} = 1 - \frac{0.0669}{V_T + 0.0213} \qquad \text{(E.7)}$$

V_T is then found by successive approximation to be 0.077 liter at STP/liter of water. Because the temperature is close to 0°C, and the cross-sectional area in the line is constant, this number is essentially the ratio of length of line with air to length of line of water required. An air lock would therefore occur if all the gas in excess of saturation in 4.5 inches/0.077=58.4 inches of pipe or in two 30-inch long sections of service line was released. Because service lines are typically 20 feet or longer and because gas released in the main may also find its way into the service line, it is entirely possible that a drop in pressure in water with a high dissolved gas content can lead to air locking in service lines.

Once pressure is raised again in the system, submarginal air locks will disappear and gradually be reabsorbed by the water traveling past them. If the volume of air trapped is great enough, however, there will still be sufficient volume even after repressurization to maintain the air lock. In such a case reabsorption by the relatively small exposed surface of water will be slower and freezing may occur before it is complete or the gas is vented.

E.5 AIR RELEASED WHEN TEMPERATURE IS RAISED

If circulation is stopped and service lines are heated with heat tape, dissolved gasses may be released. The volume the gasses occupy will be increased slightly due to the higher temperature but decreased dramatically over what they would be at atmospheric pressure if there is line pressure. With saturated water and a long enough line, it is conceivable that sufficient air for an air lock can be released. However, freezing would obviously not occur if the line is completely heat traced.

Assume the same values for C_i^*s as above. If we assume the total pressure remains at 4 atmospheres but the temperature rises to 15°C, the new concentrations at equilibrium will be:

$$C_{O_2} = 0.1363\,X_{O_2}$$
$$C_{N_2} = 0.0679\,X_{N_2} \qquad \text{(E.8)}$$

Equation E.7 then becomes:

$$X_{O_2} = \frac{0.0388}{V_T + 0.1363} = 1 - \frac{0.0669}{V_T + 0.0679} \qquad \text{(E.9)}$$

V_T is then found by successive approximation to be 0.0209 liter at STP/liter of water. Because the pressure is four times greater than at standard conditions this should be decreased by a factor of four to yield 0.0052, and because the temperature is 15°C this increases the volume slightly to 0.0055. Two 34 foot long pipes are therefore needed to supply adequate water. It is unlikely that warming a service line would produce enough gas to ever cause an air lock. In any event, such a line would likely be heat traced to such an extent that even if an air lock occurred there would not be any danger of freezing.

APPENDIX F: SERVICE LINES

Service lines are typically either nominal ¾-inch copper or 1-inch HDPE plastic. In general, copper service lines are field insulated with polyurethane foam and HDPE lines are contained within heat traced and insulated utiliducts. Advantages and disadvantages of different service line materials, sizes, and installations are discussed in the following sections and the results of freezing experiments are presented.

F.1 SERVICE LINE MATERIAL

Soft annealed copper water tube is supplied in 60 and 100 foot rolls. Copper water tube in service line sizes is normally identified by its nominal (or standard) size but the smaller sizes are also identified by the actual OD which can cause confusion at times. Type K has the thickest wall of the three types available (see Table F.1) and is the type used in service lines. While a thinner wall tube should theoretically serve as well for freeze protection because the strain that can be accommodated without rupturing is not related to wall thickness, the thicker wall tube will provide an extra margin of safety against mechanical damage and corrosion and when using electrical resistance thawing.

Table F.1: Copper Water Tube Dimensions

Nominal Size	Actual OD	Type K wall	Type K ID	Type L wall	Type L ID	Type M Wall	Type M ID
¾	0.875	0.065	0.745	0.045	0.785	0.032	0.811
1	1.125	0.065	0.995	0.050	1.025	0.035	1.055

Copper water tube manufacturing specifications are covered by ASTM Standard B 88, which require a minimum copper content (including silver) of 99.9 percent. Annealed tube larger than nominal ⅝-inch diameter must be capable of being expanded 30 percent in accordance with ASTM B 153 without visible rupture.

The tensile yield strength with 0.2 percent set for annealed copper is 33 MPa [4,800 psi] and the tensile modulus of elasticity is 125 GPa [18,000,000 psi] (Davis et al., 1982). Using these figures the strain at the tensile yield point is found to be approximately 0.03 percent. (Hard drawn copper has a strain at the yield point of approximately 0.3 percent, due mostly to its much higher tensile yield strength.)

HDPE is commonly used in the IPS pipe size in Alaska, which means the OD is the same as standard steel pipe OD of the same nominal size.

HDPE can have a tensile yield strength of 24 MPa [3,500 psi] and a tensile modulus of elasticity of 760 MPa [110,000 psi]. Using these figures the strain at the tensile yield point is found to be approximately 3 percent.

F.2 FREEZE RESISTANCE

If freezing is annular and water is bled off from the center as it is displaced, any tube material can be used without damage. If freezing is along the tube axis, and pressure is not bled off, any tube material can be made to fail. The first extreme is fairly obvious; water can be frozen in even a glass tube provided unfrozen water is allowed to escape as ice forms with no increase in pressure. It is not possible to ensure perfectly uniform freezing in a service line but it may nevertheless be approached if freezing proceeds slowly. Gilpin (1977a) did a series of tests on what he identified as ½-inch ID copper pipe. (He may have meant hard drawn copper water tube which is sometimes referred to as "pipe" although this designation should be reserved for tube which conforms to commercial standard pipe sizes (ASTM B 706). Standard ⅜-inch copper pipe has a 0.493-inch ID and a 0.109-inch wall thickness and nominal Type K ½-inch copper water tube has an 0.527-inch ID and a 0.049-inch wall thickness.) He connected both ends of a 12-foot length of the pipe to a constant pressure reservoir and installed a strain gage at the center. He insulated the pipe with ½-inch foam plastic and vigorously circulated cold air over its length to ensure uniform cooling. During most of the annular ice growth phase (which follows the dendritic ice formation phase) the strain was just that due to the water pressure from the reservoirs. The strain increased sharply just as the pipe froze completely. He reported that when the ice nucleated at $-2.5°C$ and above, the peak stress was generally less than 5,000 psi. He did not report the strain and without knowing the pipe wall thickness it can not be calculated, but even assuming he used nominal ½-inch hard drawn copper water tube the strain would still be only about 0.03 percent which is one-tenth its elastic limit. Gilpin (1977b) also reported that 1-inch ID copper pipe had a significantly lower start-up pressure than a ½-inch ID pipe for the same ice nucleation temperature. This suggests there would be less likelihood of frozen plugs forming in larger diameter tube and that ¾-inch and larger soft annealed copper water tube service lines may freeze with little or no plastic strain occurring if freezing is uniform.

Water expands 9.1 percent when it freezes. If there is uniform freezing without any pressure relief this results in 4.4 percent circumferential strain in a tube if no axial movement is permitted and about 3.5 percent circumferential strain and 1.8 percent axial strain in a tube with free axial movement, isotropic properties, relatively thin wall thickness, and plastic movement of ice.

Assuming soft annealed copper water tube can be plastically strained 30 percent without rupturing, it should be possible to freeze water in it at least six times before it fails. To test this assumption, sixteen 16-inch lengths of nominal ¾-inch Type K soft annealed copper water tube were subjected to repeated freezings. A cap was soldered on one end and a gate valve installed on the other of each tube and they were filled with water. The middle 6 inches was insulated with ½-inch foam insulation and the tubes were set horizontally on a rack outdoors in freezing temperatures. After freezing, they were thawed and water was added again. Eleven tubes failed on the fourth freeze cycle and the remaining five failed on the fifth.

The strain averaged 22 percent at the ruptures and 21 percent ½-inch from the ends of the ruptures. Assuming a tube is still serviceable after a 20 percent strain, it should be serviceable after four freeze cycles with uniform freezing and no pressure relief at the ends.

Unlike soft annealed copper, HDPE tube can withstand numerous freeze cycles because it will acquire only fairly negligible set after each cycle, but both copper and HDPE tube can be ruptured with an advancing freeze front when strains in excess of 20 percent are produced. This situation can be easily modelled by assuming negligible plastic flow of the ice and negligible change of length of a tube for a worst case analysis. Referring to Figure F.1 and noting that when a small volume of water freezes it displaces water which must be accommodated by an increase in the radius of the unfrozen length of tube, the following derivation can be done:

$$\pi r^2 \Delta x\left(1 - SG_{ice}\right) = \pi\left[\left(r + \Delta r\right)^2 - r^2\right]\left(L - x\right) \tag{F.1}$$

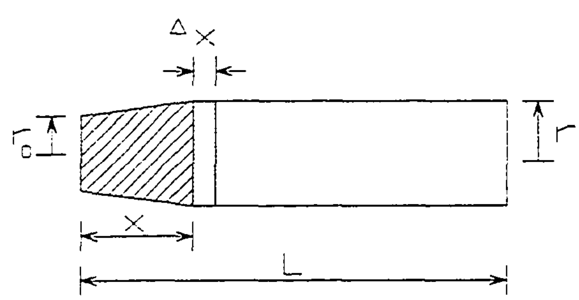


Figure F.1: Schematic of Axially Advancing Freeze-front in a Tube

$$0.0832 r^2 \Delta x \approx 2 r \Delta r\left(L - x\right) \quad when\ \Delta r \ll r \tag{F.2}$$

$$0.0416 \int \frac{dx}{\left(L - x\right)} = \int \frac{dr}{r} \tag{F.3}$$

$$-0.0416 \ln\left(L - x\right) + C = \ln r \tag{F.4}$$

$$C = \ln r_0 + 0.0416 \ln L \quad since\ x = 0\ at\ r = r_0 \tag{F.5}$$

$$\left(1 - \frac{x}{L}\right)^{-0.0416} = \frac{r}{r_0} \tag{F.6}$$

$$\frac{x}{L} = 1 - \left[\frac{r}{r_0}\right]^{-24} \qquad \textbf{(F.7)}$$

To produce 20 percent strain the freeze front would have to proceed over 98 percent of the way down the pipe. To test this, a 20-foot long piece of nominal 1-inch, 160-psi HDPE was filled with water and pushed from a 70°F room into a 0°F cold room at the rate of 16 inches every 45 minutes. Figure F.2 shows both the resulting strains and the predicted strains. Strains were calculated based on the diameter measurements of the warm pipe. As the pipe was pushed into the cold room it contracted and this accounts for the initial part having a compressive instead of a tensile strain. The tube contracted in length a small amount and ruptured before freezing was complete. These facts and some annular freezing may account for the departure of the experimental data from the predicted curve. The contraction of the tube would have been minimized if it had been pushed from a room which was maintained at just above freezing, into a room maintained at just below freezing, but this was not practical at the time and place.

The test does show that very high strains can be achieved with a progressing freeze front. With a material such as soft annealed copper or HDPE it is probably unlikely that a service line will fail due to an advancing freeze front because of the relatively long length a freeze front would have to advance.

F.3 HEAT LOSS RATES AND TIME TO FREEZE

The heat loss from uninsulated service lines exposed to air and from service lines in utiliducts exposed to air can be estimated or experimentally determined. Assume an HDPE service line with ID=0.0273 m [1.076 inches], OD=0.0334 m [1.315 inches], and k_p=0.36 W/m·K° [0.208 Btu/hr·ft·F°], a utiliduct with ID=0.10 m [4 inches], OD=0.20 m [8 inches], and k_I=0.03 W/m·K° [0.21 Btu·in/hr·ft^2·F°], still air at 223°K [−50°C; −58°F], water at 277°K [4°C; 39°F] flowing at 0.015 kg/s [0.033 lb_m/s or 0.24 gpm], and negligible thermal resistance between water and pipe or conduit and negligible radiative heat loss.

The air film coefficient, h_A, can be found from the Nusselt number, Nu, which in turn can be found from the product of the Prandtl, Pr, and Grashof, Gr, numbers. Assuming an outer pipe wall temperature, T_P, of 273°K, the average air film temperature is 248°K ≈ 250°K and the temperature drop across the air film is 50 K°. Using tabled values for air (Kays and Crawford, 1980):

$$Pr = \frac{\mu C_P}{k} = 0.724 \ at \ 250 \ °K \qquad \textbf{(F.8)}$$

For horizontal or vertical pipes and $10^4 < Gr \cdot Pr < 10^8$ (ASHRAE, 1989):

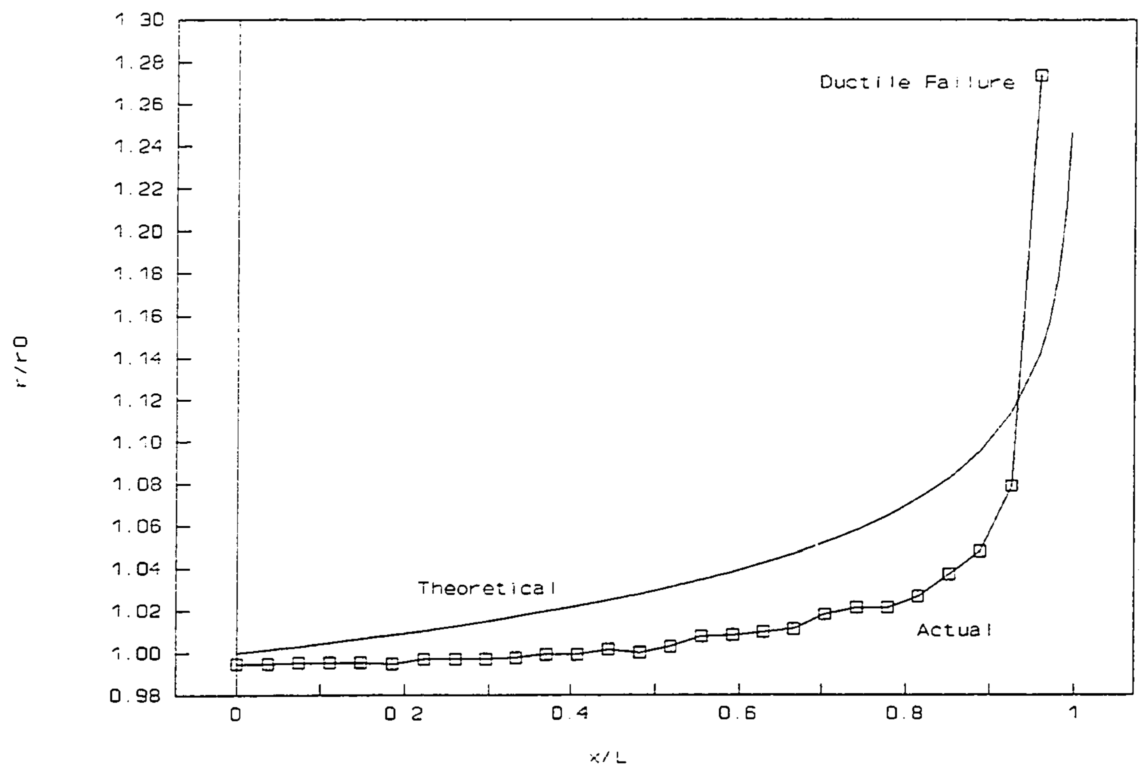


Figure F.2: Circumferential Strain in a HDPE Tube with Axial Freeze Progression

$$Gr = \frac{g\beta}{\nu^2}D^3\Delta T = \frac{(3.03\times10^8)(0.0334^3 m^3)(50 K^\circ)}{m^3\cdot K^\circ} = 5.6\times10^5 \; at \; 250 \; ^\circ K \tag{F.9}$$

$$Nu = 0.56(Pr\cdot Gr)^{0.25} = 14.2 \tag{F.10}$$

Noting that:

$$Nu = \frac{hD}{k} \tag{F.11}$$

the air film thermal resistance is:

$$R_A = \frac{1}{\pi D h_A} = \frac{1}{\pi Nu k_A} = \frac{m\cdot K^\circ}{(3.14)(14.2)(0.02226)W} = 1.01\, m\cdot K^\circ/W \tag{F.12}$$

The pipe wall thermal resistance is:

$$R_p = \frac{\ln(D/d)}{2\pi k_p} = \frac{\ln(0.0334/0.0273)}{(2)(3.14)} \frac{m \cdot K^\circ}{0.36\,W} = 0.09\,m \cdot K^\circ/W \tag{F.13}$$

The outside pipe temperature can now be calculated:

$$\frac{\Delta T}{R} = \frac{54}{1.10} = \frac{223 - T_P}{1.01} \tag{F.14}$$

$$T_P = 273\,^\circ K$$

No reiterations are needed and the heat loss for the bare service line can now be calculated:

$$\frac{Q}{L} = \frac{\Delta T}{R_T} = \frac{54\,K^\circ \cdot W}{1.10\,m \cdot K^\circ} = 56\,W/m \tag{F.15}$$

The temperature loss of the water per meter of pipe can now be calculated:

$$\frac{\Delta T_w}{L} = \frac{54\,W}{m} \frac{J}{W \cdot s} \frac{kg \cdot K^\circ}{4200\,J} \frac{s}{0.015\,kg} = 0.9\,K^\circ/m \tag{F.16}$$

Repeating the calculations for the conduit the total resistance is 4.12 m·K°/W for a heat loss rate of 13.1 W/m. Doubling the insulation increases the resistance to 6.20 m·K°/W for a heat loss rate of 8.7 W/m.

A short length of nominal one-inch Type M copper tubing was allowed to freeze. A thermistor was taped to the bottom and a second thermistor was positioned in the air about 4 inches above one end of the tubing. Figure F.3 shows the ambient temperature and cool-down of the tubing. The mass of water per meter is:

$$\frac{(0.0268^2 m^2)(3.14)}{4} \frac{10^3 kg}{m^3} = 0.564\,kg/m \tag{F.17}$$

The initial cooling from 5 °C to -5 °C takes about 10 minutes so the heat loss rate per meter is approximately:

$$\frac{Q}{L} = \frac{0.564\,kg}{m} \frac{4200\,W \cdot s}{kg K^\circ} \frac{1\,K^\circ}{60\,s} = 39\,W/m \tag{F.18}$$

The freezing takes about 70 minutes so the heat loss rate per meter is approximately:

$$\frac{Q}{L} = \frac{0.564\,kg}{m} \frac{3.334 \times 10^5\,W \cdot s}{4200\,s \cdot kg} = 45\,W/m \tag{F.19}$$

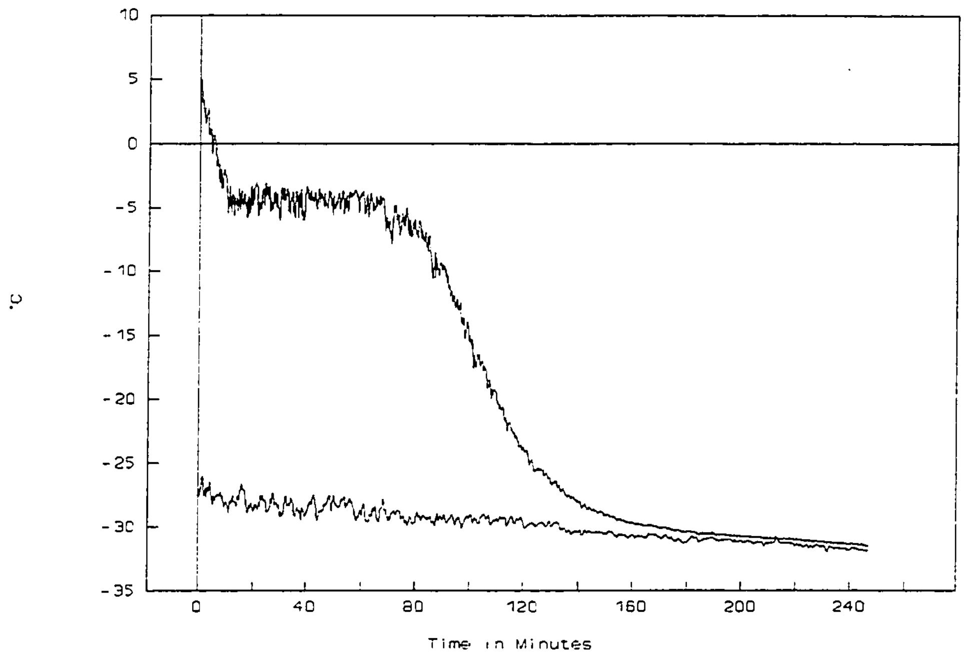


Figure F.3: Air and Surface Temperatures of One-inch Copper Tubing During Freezing

The theoretical heat loss is 17.5 W/m assuming the air is still, the water film resistance is negligible, there are no radiative heat losses, and that the outer pipe wall temperature is given by the thermistor on the pipe.

With these high rates of heat loss it is clear that every effort should be made to ensure an uninsulated service line is never exposed to cold air.

F.4 THAWING

Copper tube can be thawed by passing an electrical current through it. Amsbary (1936), Currey (1980), Nelson (1976), and Sheppard (1934) discuss this method of thawing service lines. This type of thawing must be done with low voltage for safety reasons. Short circuits are a problem and piping must be isolated before thawing is attempted. There is a possibility of electrolytic corrosion also. Uniprise International Inc. (Terryville, Connecticut) manufactures a device specifically designed for electrically thawing pipes.

Electrical thawing requires good conductivity at the pipe joints. Copper service tube should be continuous or joined with flare type couplings; soldering is not acceptable because thawing may result in leaks; compression fittings are not acceptable because conductivity is not ensured. The tubes should be electrically isolated along the length and securely jumpered at the main. Continuity testing is recommended before backfilling.

Currey and Nelson also describe the use of hot water to thaw service lines from the inside.

Western Water Ltd. (Calgary and Edmonton, Alberta) markets a device for interior thawing service of lines. Heat trace may also be used for thawing.

F.5 SERVICE LINE CONNECTIONS TO THE MAIN

Service lines may be connected to a main with corporation stops or pitorifices threaded directly into a ductile iron (DI) main or threaded into saddles on a main. If the main is HDPE the saddles should be stainless steel wrap around type with honeycomb rubber to ensure a good seal even with thermal contraction or stress relaxation in the HDPE main.

Copper water tube should be connected to the corporation stops with flare fittings. HDPE tube is normally connected with compression fittings.

Fusion welding may be used if both the main and the service is HDPE. This can result in a very dependable joint if proper care is taken. Welders must be properly certified. It is very easy to make joints that will look fine and which will withstand initial pressure testing, but are incomplete and which will fail in time.

F.6 CONCLUSIONS

Soft annealed copper water tube with foamed in place polyurethane insulation is the cheapest way to install a service line. It will probably withstand at least three freeze-thaw cycles without damage and may survive an indefinite number of cycles. Copper is recommended where freezing is not likely.

HDPE will survive numerous freeze-thaw cycles but it will need to be heat traced or provisions must be made for internal thawing. If heat trace is used, provision should be made for replacing it. Heat trace does not allow quick thawing and it can fail. Heat trace is also expensive. Special attention must also be given to transitioning from HDPE. HDPE is recommended for situations where repeated freezing is possible.

APPENDIX G: PITORIFICES AND SMALL PUMPS

This appendix is written as a stand-alone summary of the thesis "Pitorifices and Small Pumps in Cold Region Water Distribution Systems" by Michael Mauser submitted to the faculty of the University of Alaska Fairbanks on May, 1995.

A study to improve the energy efficiency of cold region potable water distribution systems was conducted. During the course of the study nine different pitorifice shapes were tested in full and model scale and small circulation pumps from ten different manufacturers were evaluated. Field tests on both pitorifices and pumps were done in two remote Alaskan communities and in Fairbanks. This appendix focusses on what was learned that can be directly applied to the design of cold region water distribution systems.

G.1 PITORIFICES

Pitorifices are scoops which project into the water main. They are used to divert flow from the main to the house and back to the main through dual service lines so that water in the service lines is constantly replaced by warmer water and freezing is prevented. The first pitorifices were made in 1953 by bending a copper tube, trimming it, and turning it in a lathe so that it had a cylindrical profile. The tube was then soldered into a corporation stop which was screwed directly into a wood stave main. Over the years a number of variations have been used; the six most common are shown in Figure G.1, grouped into three categories.

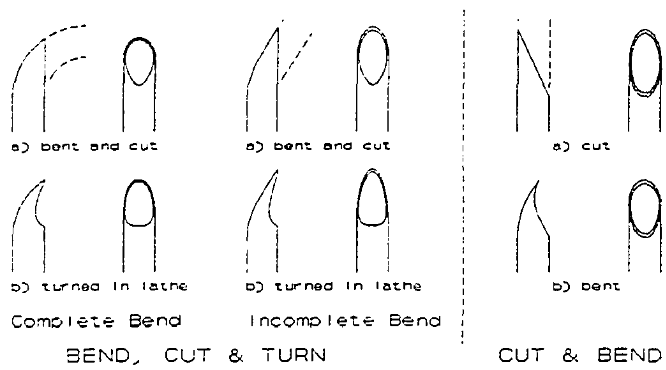


Figure G.1: Six Common Pitorifice Shapes

Mueller (Decatur, Illinois) supplied the original complete bend, cut and turn type pitorifices but no longer manufactures pitorifices. Ford Meter Box (Wabash, Indiana) manufactures incomplete bend type pitorifices. Western Utilities Supply (Seattle, Washington) supplies pitorifices made using cut and bent pieces of copper tube and Mueller corporation stops. Of the six variations shown in Figure G.1, all provide roughly equivalent flows with the exception of the one made by simply cutting the end at an angle which performed relatively poorly. The best performing pitorifice was found to be the incomplete bend and cut type. This type of pitorifice can also be made by soldering a 45° fitting ell on a copper tube and trimming it with a metal cut-off saw. A corporation stop with AWWA thread is recommended over one with NPT because it will accommodate a larger pitorifice tube; in the case of a 1-inch corporation stop a nominal 1-inch diameter copper water tube is accommodated which in turn allows the use of a standard size 45° fitting ell in the manufacture. The added costs associated with using nominal 1-inch rather than ¾-inch diameter pitorifices and service lines will often be offset by savings in pumping costs over the life of the system.

Performance of incomplete bend and cut type pitorifices as a function of position and relative size is shown in Figure G.2. The degree of insertion is indicated as the ratio of the distance the tip is from the center of the main to the radius of the main. The pitorifices were made with 45° ells soldered to copper or brass tubes with outside diameters (d) of 0.5, 0.625, 0.875, and 1.125 inches inserted into PVC mains with inside diameters (D) of 4.1 and 6.1 inches. The pitorifices were inserted 6-inches apart on center and oriented to face directly up and downstream. Increasing the spacing will result in slightly better performance, but the increase is not significant. Changing the orientation within 5° does not affect performance.

The head difference developed across a pitorifice pair (ΔH_{po}) is primarily due in about equal parts to the dynamic pressure developed at the upstream pitorifice and the pressure drag at the downstream pitorifice. The head difference is given by:

$$\Delta H_{po} = \frac{K_{po} V_m^2}{2g} \tag{G.1}$$

where K_{po} is the pitorifice coefficient, V_m is the velocity in the main, and g is the acceleration of gravity. K_{po} increases slightly as the ratio of the pitorifice outside diameter to the main inside diameter (d/D) increases and as pitorifice is inserted deeper into the main because of the increasing contribution of the head loss across the pitorifices and because of the increase in the velocity resulting from the decrease in cross-sectional flow area.

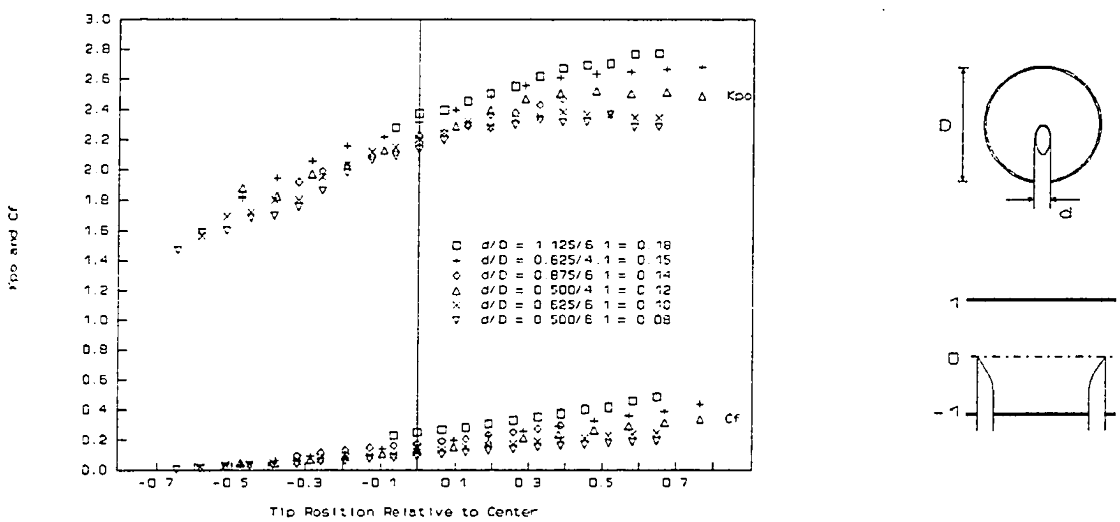


Figure G.2: Performance of Incomplete Bend and Cut Type Pitorifices

When designing a water distribution system a minimum service line mass flow rate ($\dot{m}_s$) is determined based on the estimated heat loss rate ($\dot{E}_s$) of the longest service line:

$$\dot{m}_s = \frac{\dot{E}_s}{\left(T_R - T_{min}\right) C_p} \tag{G.2}$$

where T_R is the return temperature maintained in the main, T_{min} is the minimum allowable temperature in the service line (usually several degrees above freezing) and C_p is the specific heat of water. The required velocity in the service line (V_s) is then given by:

$$V_s = \frac{4 \dot{m}_s}{\rho \pi D_s^2} = \frac{4 \dot{E}_s}{\rho \pi D_s^2 \left(T_R - T_{min}\right) C_p} \tag{G.3}$$

where ρ is the density of water and D_s is the inside diameter of the service line.

During laminar flow (which is the usual condition) the head loss in the service line (ΔH_s) is solved as:

$$\Delta H_s = f \frac{L_s}{D_s} \frac{V_s^2}{2g} = \frac{64\ \nu}{D_s V_s} \frac{L_s}{D_s} \frac{V_s}{2g} \frac{4 \dot{E}_s}{\rho \pi D_s^2 \left(T_R - T_{min}\right) C_p} = \frac{128\ \nu L_s \dot{E}_s}{D_s^4 g \rho \pi \left(T_R - T_{min}\right) C_p} \tag{G.4}$$

where f is the Darcy friction factor, ν is the kinematic viscosity of water (use the value at the freezing point for design purposes), and L_s is the total equivalent length of service line which is the total length of tubing for the longest service plus about 30 feet equivalent length for losses in the pitorifices and about 10 feet equivalent length for losses in the fittings at the service. (Equivalent lengths for laminar flow are high compared to those used for fitting losses in turbulent flow.)

The required main flow velocity (V_m) can be approximated by combining Equations G.1 and G.4:

$$V_m = \left(\frac{256\, \nu L_s \dot{E}_s}{D_s^4 K_{po}\, \rho\, \pi \left(T_R - T_{min}\right) C_p} \right)^{\frac{1}{2}} \qquad \textbf{(G.5)}$$

Because pumping costs vary roughly with the velocity cubed, using larger diameter service lines, using more insulation to decrease heat loss, adding heat to increase the return temperature, and using individual circulation pumps on the longest services to reduce the total equivalent length of the longest service served by pitorifice can all contribute to lower operating costs. It is the design engineer's responsibility to determine an optimum mix that will allow minimizing operating costs while ensuring reliability by maintaining a reasonable minimum temperature. Generally, flow rates and return temperatures are maintained fairly constant over the entire operating season. Two speed pumps are sometimes used so a higher pumping rate is possible during the coldest periods. Alternatively, if supplemental heat is added, a higher return temperature can be maintained at these times.

The head loss in the main due to the presence of the pitorifice pair is fairly minimal. It can be estimated by:

$$\Delta H = \frac{C_f V_m^2}{2g} \qquad \textbf{(G.6)}$$

where C_f is the head loss coefficient for the pitorifice pair. Figure G.2 shows C_f increases as the pitorifices are inserted further into the main but so does K_{po} which allows the velocity in the main to be dropped. Dropping the velocity in the main means all the head losses in the entire length of main are reduced, and this will tend to offset the additional head loss from the pitorifices. Inserting the pitorifices to the center line of the main is recommended for 6-inch mains.

G.2 TESTING PITORIFICE PERFORMANCE AT SERVICES

Testing pitorifice performance has proved useful in identifying problems with circulation in the main, with damaged or incorrectly installed pitorifices, or with high heat loss. The correct hydraulic functioning of the pitorifices can be checked by measuring the shut-off head across the pair with a manometer and using Equation G.1. Manometer water levels will oscillate making an exact reading difficult at times. The magnitude of the oscillations will increase with increasing flow rates in the main and the frequency will increase with shorter service line lengths. Variations in the flow rate in the main will also cause oscillations. In water systems with no large water users and little or no branching oscillations are not usually a problem.

The shut-off head measurement will show that there is flow in the main and a pressure differential across the service line, but it doesn't indicate if there is adequate flow in the service line. The flow rate in the service line can be estimated from the shut-off head and the total service line length using Equation G.4, but this presumes undamaged pitorifices and no kinking, plugging, or deposits in the line. Where manometer water levels are fairly stable and the shut-off head is 1-inch or more, service line flows can be measured with a bypass manometer. A bypass manometer is simply a manometer with a valve between the two legs which can be opened to allow some flow. A small ball valve can be used, or a large ball valve with a restriction either in the valve itself or in the piping between the manometer legs. By experimentally determining the head loss across the bypass as a function of flow, a system curve can be drawn. Measuring both the shut-off head and the bypass head allows extrapolation to the flow that would be present without the restriction. Figure G.3 shows how a bypass manometer is used and Figure G.4 shows the determination of a service line flow by extrapolating from the shut-off head through the bypass head. The system curve shown was determined for a 1-inch ball valve which had a 0.5-inch long plug with a 0.25-inch diameter hole.

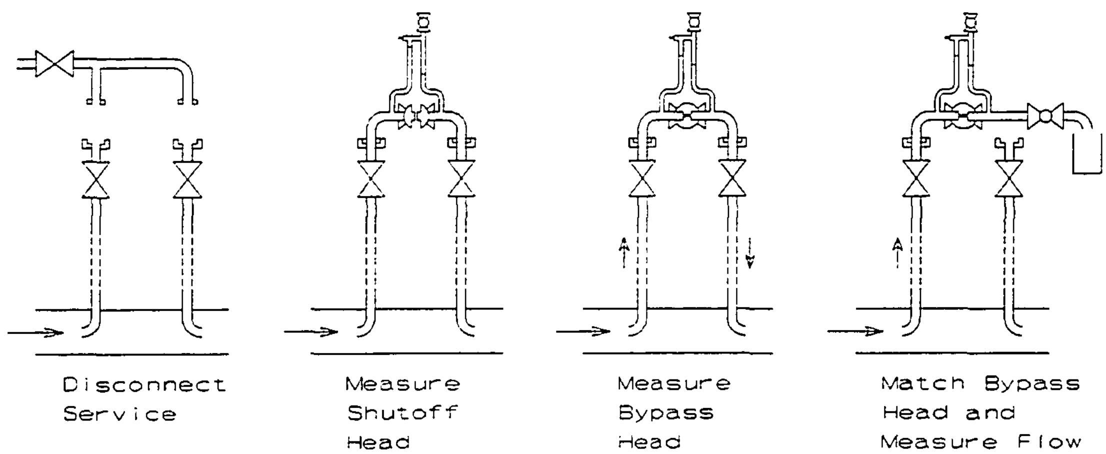


Figure G.3: Bypass Manometer Used to Test Service Line

Instead of using the calibration curve, the flow can also be found by withdrawing water off one side or the other through a throttling valve and determining the flow, Q_{BP}, which corresponds to the previously measured H_{BP}. The unrestricted flow can now be estimated using the linear relationship $Q \approx H_{SO}Q_{BP}/(H_{SO}-H_{BP})$.

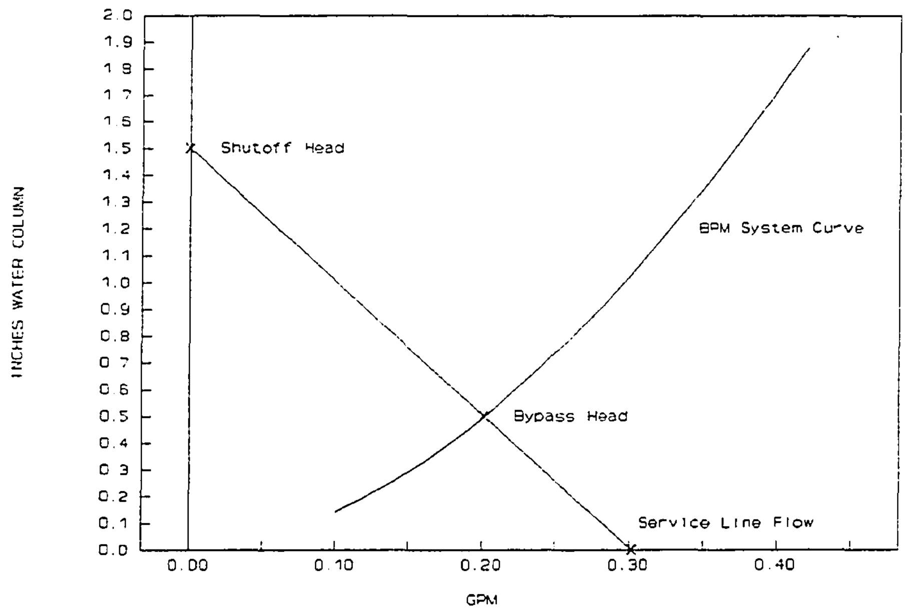


Figure G.4: Bypass Manometer (BPM) System Curve with Service Line Plot

In the Fairbanks area where service lines enter a heated space below grade, a minimum flow of 0.2 gpm for every 100 feet of distance from the main seems to be acceptable. Where ground temperatures are colder or where the service lines enter the house above grade, higher flows may be desired.

A much more accurate flow measurement can be made using a low head loss flow meter. An ISS Clorius International (Ballerup, Denmark) model 3VPD magnetic flow meter has a range from 0.01 to 13 gpm with negligible head loss and it costs less than $2,000. An ultrasonic transit time flow meter might also be suitable. Controlotron (Hauppauge, New York) sells one for about $4,300 with about the same range as the Clorius magnetic flow meter. The Controlotron flow computer can be field modified for use with optional strap on transducers to measure flows in larger diameter pipe which expands its upper range considerably. No other types of flow meters were found with low enough head loss and sufficient accuracy to be readily useful.

The actual heat loss in a service line can be estimated by installing a flow meter and a thermistor on the service loop at the house. The temperature in the main can be measured by withdrawing water at a high rate from a tap. Combining this information with a flow and temperature reading taken with no water use after thermal equilibrium is reached allows the heat loss for one half the leg to be estimated. Measurements should be taken when heat loss is at a maximum, and even then the flow may have to be throttled to produce a significant temperature drop. The upstream pipe in the house should be insulated during the test.

G.3 INDIVIDUAL CIRCULATION PUMPS

The flow required for freeze protection is determined using Equation G.2. Flows are usually low so a low flow, low head pump is appropriate. Such pumps are generally intended for hydronic heating systems, solar heat collection systems, potable water heat exchange systems, or potable hot water circulation systems. They have either a magnetic coupling between the pump impeller and the motor or a motor rotor which is connected directly to the pump impeller and lubricated by the system water. Both the magnetic coupling and the wet rotor (also called system lubricated) designs minimize problems with seals and bearings. In general, the magnetic drive pumps loose most of their energy to the atmosphere and the wet rotor pumps loose most of their energy to the water being pumped. Only 2 to 20 percent of the energy goes towards pumping the water.

Only pumps suitable for potable water which used either AC or brushless DC motors were considered. Because electric rates can exceed $0.40/kWh, only pumps using less than 50 watts ($175/yr at that rate) were considered. The only pumps located meeting these conditions were manufactured by Grundfos (Clovis, California), Hartell (Ivyland, Pennsylvania), Laing Thermotech (Chula Vista, California), Little Giant (Oklahoma City, Oklahoma), and March Manufacturing (Glenview, Illinois). All the pumps come with ½-inch fittings, the Hartell pumps can be modified to accept ¾-inch sweat copper fittings, but only the Grundfos pump has ¾-inch fittings standard. Ivan Labs (Jupiter, Florida) makes a high and low amperage brushless DC driver for the March pump. Information on the pumps is summarized in Table G.1 and the pump curves are shown with representative system curves in Figure G.5. The lower amperage Ivan Labs driver is shown, their higher amperage model produces about 15 percent more head. All the pumps with brushless DC motors or drivers can be run at lower voltages with less wattage required and less performance and they all require a DC power supply which may be 50 to 80 percent efficient.

When cold water is drawn through a wet rotor pump which is not operating, atmospheric water vapor may condense on the cooled windings and cause damage. Laing hermetically seals their pump and the Ivan Labs driver windings are impregnated with a potting compound. Grundfos used a potting compound in the past, but now they seal the windings in the housing with a rubber plug to prevent condensation. This apparently is sufficient, but the pump warrantee does not apply for use with water below 50°F. The Hartell pump windings are exposed to the atmosphere so it is probably much more at risk to damage. If the Hartell wet rotor pump is used it should be operated continuously because there can often be high humidity at the pump location.

Figure G.6 shows the different ways circulation pumps can be plumbed. In the first, water to the customer is taken off downstream of the pump and the flow through the pump will exceed the open discharge flow of the pump when water is withdrawn at twice the open discharge flow rate (assuming both

Table G.1: Pump Information

Pump Model	Wattage	Type	Notes
A. Grundfos UM15-10B	38	Wet Rotor	
B. Hartell WR	50	Wet Rotor	1
C. Hartell MD	30	Magnetic Drive	2
D. Hartell HEL	10	Magnetic Drive	3
E. Hartell HEH	21	Magnetic Drive	4
F. Laing 303	33	Wet Rotor	5
G. Little Giant CMD-100-3B	34	Magnetic Drive	2
H. March 809	30	Magnetic Drive	2
I. March 809/Ivan Labs	4	Wet Rotor/Magnetic Drive	6

1. Unsealed windings may be susceptible to condensation.
2. Should be plumbed upstream of user tap unless check valve used to prevent backflow.
3. Brushless DC motor with 17 V operation. Power supply losses are not included.
4. Brushless DC motor with 14 V operation. Power supply losses are not included.
5. Should be plumbed downstream of user tap with check valve to prevent backflow.
6. Brushless DC driver with 17 V operation. Power supply losses are not included.

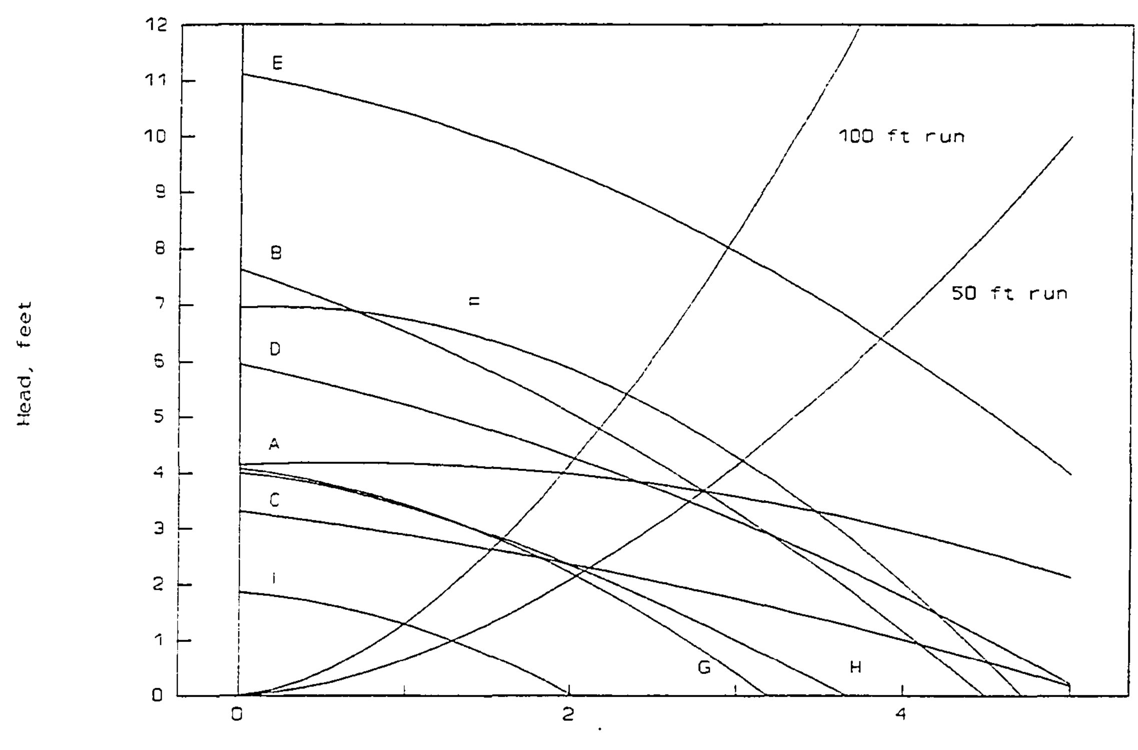


Figure G.5: Pump Curves and ¾-inch Service Line System Curves

legs have the same ID). This is because at open discharge the pump contributes no head and both legs of the dual service line would have the same flow. This can lead to overloading the motor in a large pump, this was not found to be a problem with these small pumps. Laing recommends against installing their pump in this location, but this is probably because its unique construction makes it susceptible to vibration damage.

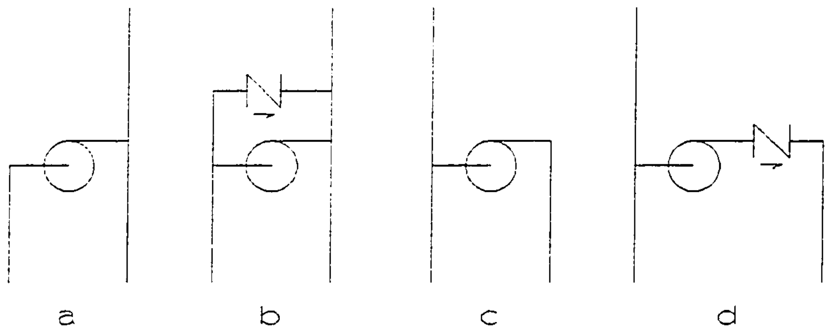


Figure G.6: Pump Plumbing Arrangements

A check valve can be added to allow water to bypass the pump when the withdrawal rate exceeds twice the pump open discharge flow rate as shown in Figure G.6b. However, if this check valve were to fail open, there would be no circulation in the service line.

Figure G.6c shows water taken off upstream of the pump. In this configuration flow through the pump may be stopped or reversed when water is withdrawn. Laing recommends installing a check valve with the pump to prevent this forced backflow. In the case of the Laing pump, a backflow causes vibration. In all the pumps, backflow causes a higher electric current flow. The longer the service line, the smaller the diameter of the service line, or the lower the pump shut-off head, the easier it is for water to be drawn backwards through a small circulation pump. For example, a pump with a shut-off head of 4 ft installed on 100 ft run of ¾-inch service line will be pumping at shut-off if 2 gpm is being withdrawn and will have water drawn through it backwards at flows in excess of that as can be seen by referring to Figure G.5. A check valve can be added in series with the pump to prevent back flow as shown in Figure G.6d.

If the service also uses pitorifices, the use of check valves should be avoided. Even if the check valve were oriented to allow flow from the pitorifices in the event of a pump failure, the head loss caused by the check valve would prevent adequate flow in most cases. (Head loss through an inactive pump is relatively minor.)

When water is forced backwards through a magnetic drive pump driven by a motor which has a higher torque than the magnetic coupling, the drive may decouple. Once decoupled it may fail to recouple even after water flow has stopped, and while decoupled the impeller assembly may vibrate and cause damage. A check valve will prevent backflow. The brushless DC motor used by the Hartell HEL pump has a lower torque than the magnetic coupling and would not require a check valve.

G.4 FLOW ALARMS AND SWITCHES

Flow switches are occasionally used with individual circulation pumps to trigger an alarm or turn on a heat trace. The benefits and costs of using a flow switch should be carefully considered. If freeze-ups are relatively uncommon or relatively easy to deal with, the added costs of a flow switch may not be justified. The total cost of a flow switch is not just the purchase price (usually well over $100) but the installation cost and operating cost as well. The operating cost can be relatively expensive if the flow switch requires flows higher than would otherwise be required to function or if it induces a high head loss requiring a larger pump. In such cases a more expensive model flow switch may be justified. To illustrate this point consider a Grundfos model UP15-42SU 85 watt pump used with an Imo Industries (Plainville, Connecticut) Gems AFS-400 Series A26440 shuttle path flow switch. The pump performance and the flow switch head loss curves are shown in Figure G.7. Also shown is the equivalent pump performance curve obtained by subtracting the flow switch losses from the original pump curve.

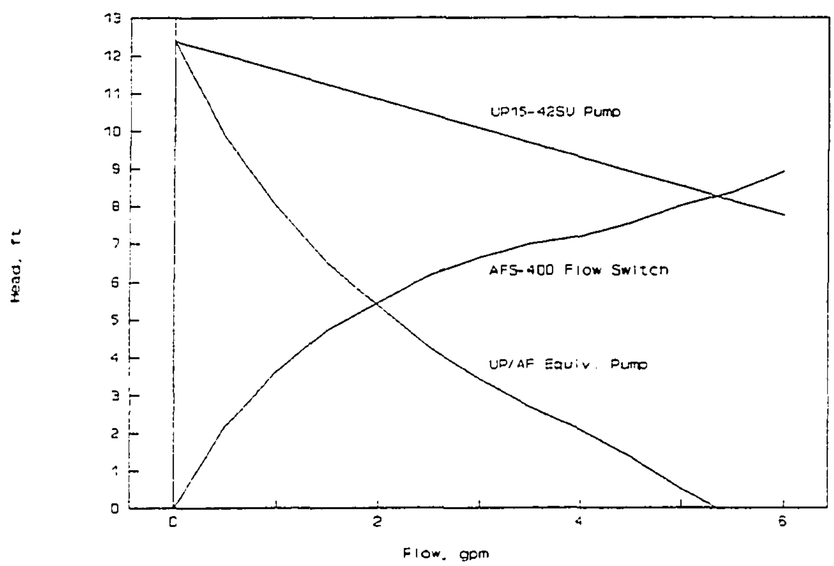


Figure G.7: Grundfos UP15-42SU Pump with Gems Flow Switch

Figure G.8 shows the performance curve of a Grundfos model UM15-10B 38 watt pump, the head loss curve of a McDonnell & Miller (Chicago, Illinois) FS6 Series bypass flow switch, and the resulting equivalent pump curve.

Figure G.9 shows the two equivalent pump curves with system curves for a 400 foot long, 1-inch service and a 100 foot long, ¾-inch service. For shorter service lines the resulting flow rate will be the

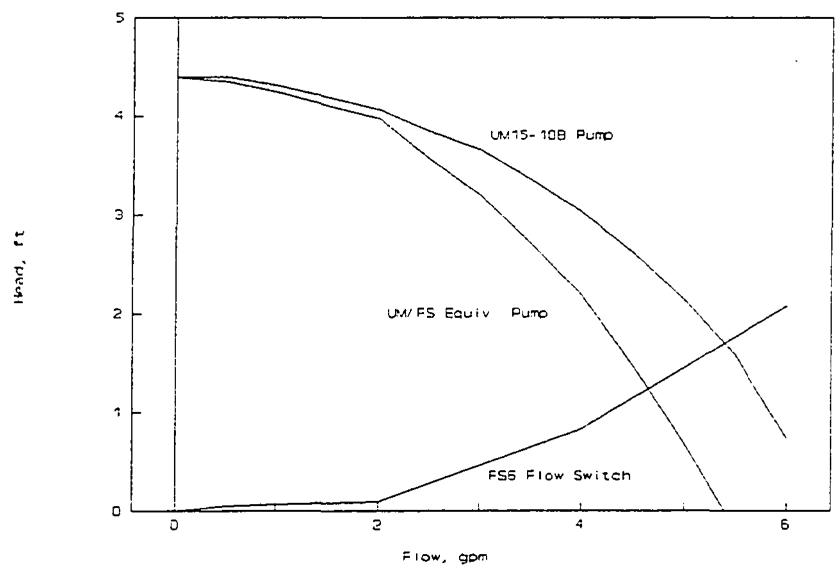


Figure G.8: Grundfos UM15-10B Pump with McDonnell & Miller Flow Switch

same but the larger pump would cost over $160 more a year for electricity to operate where electric costs are over $0.40/kWh. If heat addition to the water is required some credit should be given for the reduced heating cost; assuming 80 percent of the electric energy ultimately goes to heat the water and that heat is available at one third of the cost of electricity, the net additional cost of the larger pump would be about $120 per year.

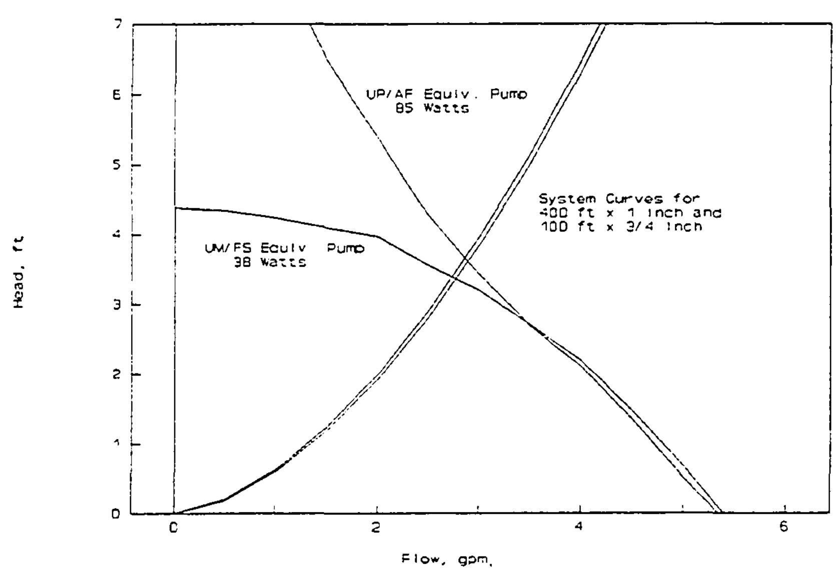


Figure G.9: Equivalent Pump Performance and System Curves

G.5 PITORIFICES VERSUS INDIVIDUAL CIRCULATION PUMPS

In general, using a small circulation pump at each house instead of pitorifices is not recommended. The initial cost will be greater, reliability less, replacements more frequent, and the total operating costs will usually be higher. The Birch Estates Loop in Fairbanks provides a good example of this. The loop serves 66 residences. A 2 kW pump circulates water through about 4,000 feet of 6-inch main for a seasonal operating cost of about $900. A much smaller pump could be used on the main if every house had a small service line pump. However, assuming 40 watt pumps at each house, the total cost of operating the 66 pumps would be about $1,200 assuming everyone remembered to turn them off every season or $2,400 if everyone left them run year round. Year round operation is frequently advised, particularly where deposits may form during long periods of inactivity which could resist start-up torque.

Where service line lengths are very great and where there are very few service lines on a main, it may be advisable to use individual circulation pumps. In such cases there may still be some advantages to installing pitorifices in addition to the circulation pumps. The added cost of the pitorifice is relatively small and pitorifices can provide some circulation in the service lines even with long service lines or low circulation rates in the mains. This flow may be enough in many cases to prevent freeze-ups that would otherwise occur. Also, as additional services are added and the operating economics change, the use of individual circulation pumps may be phased out.

The presence of pitorifices in a main can hamper internal steam thawing efforts and pitorifices have been damaged by debris or ice in the mains leading to leaks but these are relatively infrequent problems.

G.6 OTHER CONCERNS

The heat gains from pumping can be significant whether pitorifices or individual circulation pumps are used. If this heat would otherwise have to be added to the water in the loop then the pumping cost should be adjusted down to reflect the value of this heat. Even if the water in the loop does not require this heat for freeze protection, the heat may be considered of some value to the consumer because the warmer the water is supplied, the less additional heating is required by the consumer. Despite this, it is almost always better to conserve electricity and to use the most efficient motors and pumps available because electricity is usually more expensive than heat. The net savings will usually justify the additional expense of more efficient equipment. For example, a high efficiency 2 hp motor may cost only about $60 more than a standard model but can save over $190 in electricity in 6 months where electricity costs $0.40/kWh. If the pump house must be heated and if the value of heat is one third that for electricity, the net savings would be over $120 per season.

It may be advisable to use a larger motor and pump for the main than the minimum size. Electric motors have their highest efficiency at somewhat less than full load, plus they operate cooler and will have a longer life at lower loads. Coupled to an oversized pump with a suitably trimmed impeller future expansion of the loop may be accommodated by just changing the pump impeller; also, the wire to water efficiency of such a motor and pump combination may actually be quite high.

The Hazen-Williams formula is frequently used by engineers to calculate head loss in piping. While it is fairly accurate it should be kept in mind that the C factor for a pipe needs to be adjusted when it is used for conditions significantly different than the test conditions under which it was determined (usually 3 fps and room temperature). If $C=140$ for 3 fps and 68°F, it will be 135 for 1.5 fps and 40°F with most of the correction due to the higher viscosity of water at the lower temperature.

The use of high density polyethylene (HDPE) has become very popular for water mains. Connections for services can be made with fusion welded tees or saddle type taps. Certification of welders under actual field conditions by destructive testing of welds is strongly recommended. The use of spring washers and tapped, wrap-around stainless steel repair clamps such as supplied by Romac Industries (Seattle, Washington) is recommended. The spring washers and the waffle gasket accommodate the increased deformation and thermal contraction which can occur in HDPE.

G.7 CONCLUSIONS

With a better understanding of pitorifices it is possible to depart with confidence from the "rule of thumb" which dictated 2 fps velocities in mains and a 60 foot limit to service lines and make dramatic reductions in the energy requirements for pumping. The design engineer should determine the mass flow needed for the expected heat loss and the allowable temperature drop and calculate the required flow in the main which provides that flow. Investments in 1-inch or even larger diameter service lines will be paid back in a few years in energy savings in most cases.

An incomplete bend and cut type pitorifice is recommended and insertion of the tip should be to the center line of the main when practical.

Often the use of small circulation pumps on the longest service lines can allow additional savings by lowing the required flow in the main. In most situations a 40 watt or smaller pump will suffice.

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